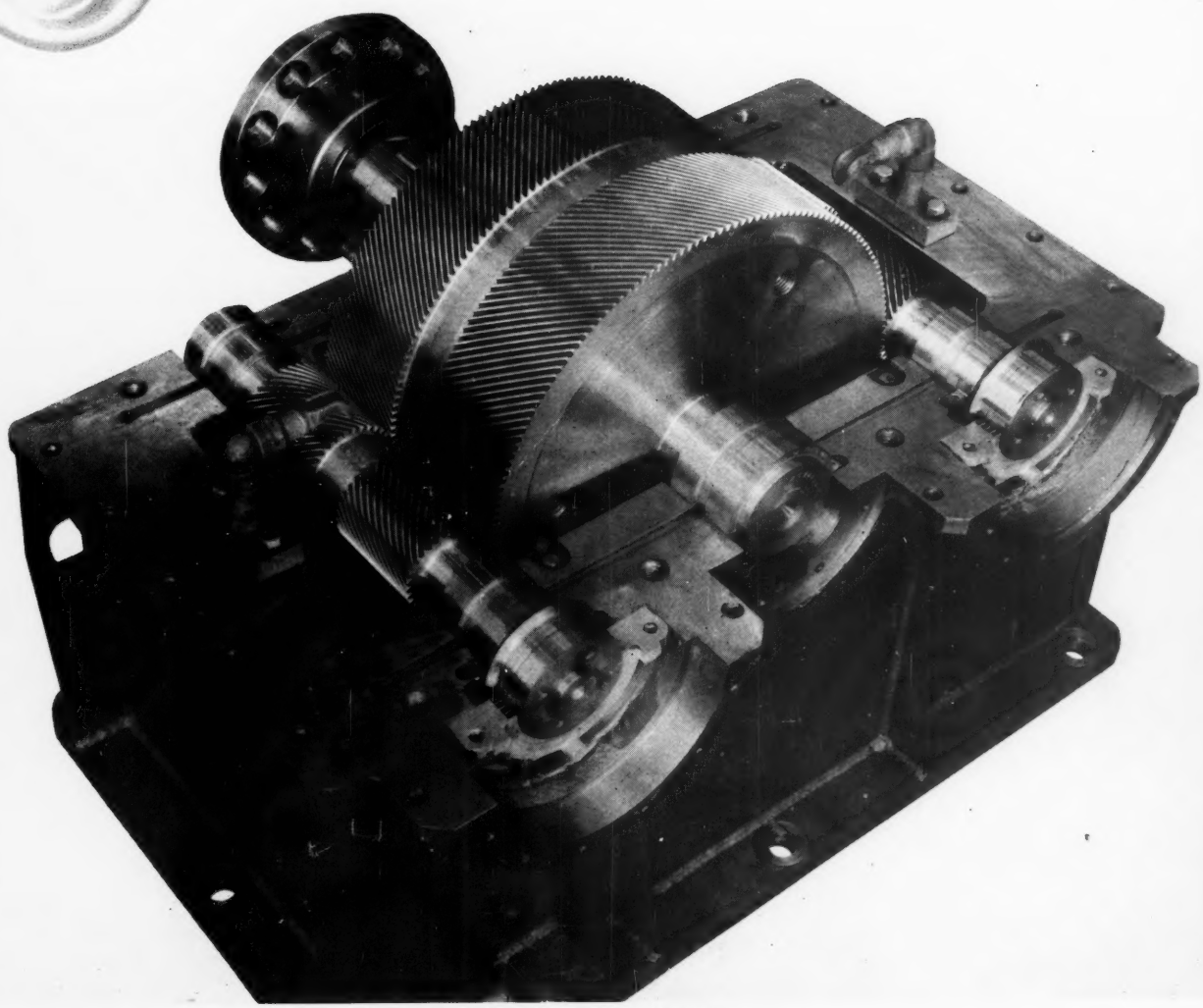
Practical Feature: LUBRICATION REQUIREMENTS OF GEARS AS SEEN BY A GEAR ENGINEER

Lubrication ENGINEERING

OCTOBER, 1953
Vol. 9, No. 5



SPEED INCREASING GEAR

JOURNAL OF
THE AMERICAN SOCIETY OF LUBRICATION ENGINEERS

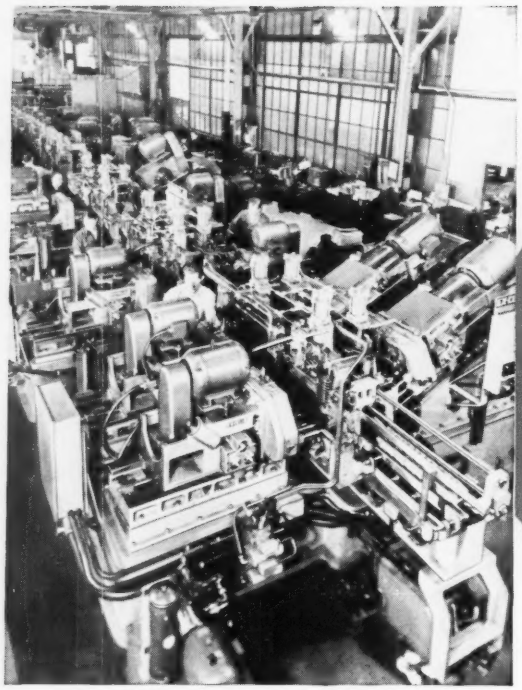


Photo Courtesy Ex-Cell-O Corp., Detroit, Mich.

Trabon

AUTOMATIC LUBRICATION
used on transfer machines
in large automotive plant

this has a BEARING on the subject

Illustrated here are transfer machines typical of several installed in the manufacturing plant of a large automotive producer . . . the first of which was put in operation during the early part of 1951.

Each of these machines is being automatically lubricated with a TRABON system. Two systems cover 195 bearing points each; two, 151 points each; and two, 305 points each.

Each system is so proportioned that slides, ways and heavily loaded working bearings receive the proper amount of lubricant every 20 to 30 minutes. All minor bearings on fixtures receive the exact amount of lubricant required every four hours.

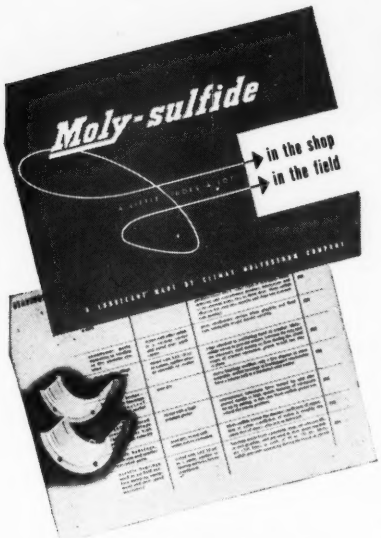
Whether your problem is to lubricate a hot metal crane, crushers, ore unloaders, presses, rolling mills, blast furnaces, boring mills, or any other industrial application . . . a TRABON system is your answer.

Completely sealed—positive—hydraulic—handles oil or grease—backed by years of experience.

WRITE FOR BULLETIN 529

Trabon ENGINEERING CORPORATION
1814 E. 40th STREET • CLEVELAND 3, OHIO
Oil and Grease Systems

154 ideas on ways to use...



154 varied applications of molybdenum sulfide in the shop and in the field are described in a new booklet now available. This solid-film lubricant has demonstrated unique anti-friction properties under conditions of extreme pressure, high velocity, elevated temperature, or chemical attack.

The 40-page booklet contains the records of solved lubrication problems — some might solve your own.

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Position.....
Company.....
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MOLY

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Lubrication ENGINEERING

OCTOBER, 1953, Vol. 9, No. 5

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(Cover Picture courtesy of General Electric Co.)

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JOURNAL OF
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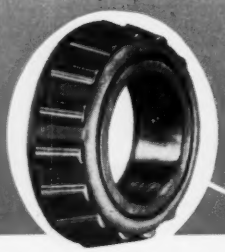
give BENTONE*GREASES your tough lubrication jobs...



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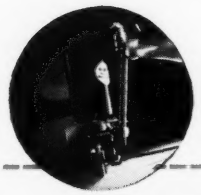

developed... **for "tough-job" lubrication on the largest, heaviest machines...**

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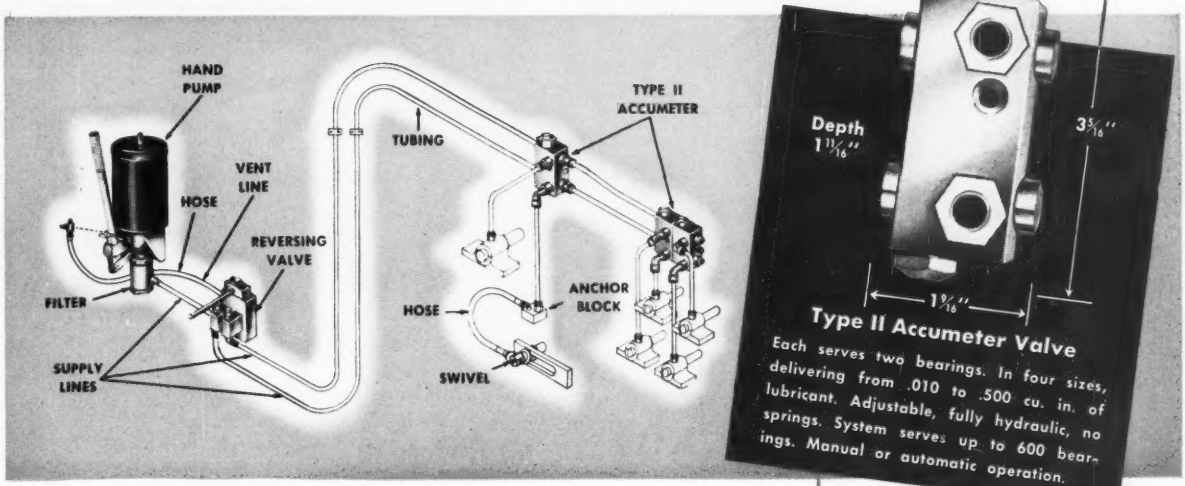
sealed against grit...fumes...abrasives
operates outdoors in any weather!



Alemite Accumeter Lubrication System delivers the exact, measured amount of lubricant to *all* bearings of a machine. In a fraction of the time required for hand lubrication—Accumeter measures and delivers lubricant—*while the machine is in operation!* No down time—no points missed. No wonder 95% of big plants buying machine tools want centralized lubrication!

Type II Accumeter is unmatched for simplicity of design. It's fully sealed and enclosed—yet may be serviced without being removed from machine. Operates in salt spray, severe acid or fume conditions—even totally immersed—because it may be painted or treated with non-corrosives.

Type II Accumeter System can serve single machines or groups—will handle any required lubricant—heavy or light. It is just one of three types of Accumeter Systems made by Alemite. One of them will serve your requirements.

offers...

ALL THESE ADVANTAGES!

- **Eliminates shutdown time for lubrication. *Adds* productive time to machine output.**
- **Seals completely against dirt, grit, water all the way from "Barrel-to-Bearing."**
- **Prevents bearing troubles due to neglect or use of wrong lubricant.**
- **Services *all* bearings—including those inaccessible or dangerous—in one operation.**
- **Avoids work spoilage and bearing repairs due to over-lubrication.**

FACTORY-TESTED . . . FIELD-PROVED

Proved in the field. Exhaustive tests showed *no variation* in the amount of lubricant discharged . . . even after 73,312 lubrication cycles, *equal to 122 years of twice-a-day service.*

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. . . FED UNDER PRESSURE
BY THE MEASURED DROP

the most dependable method of lubrication ever developed!

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There are six different models to meet almost every application requirement. Please write us for all details regarding your particular lubricator requirements.

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• Skilled in Die Casting Mechanics • Experienced in Lubrication Engineering • Originators of Really High Speed Air Tools

...with TYCOL lubricants on hand!

A Cutting Oil with a "Delicate" Touch! Screw threading to fine tolerances is the acid test of any cutting oil . . . one reason why we ask: can your present oil measure up to Tycol Angrove 40? Take the customer who was threading 3/8" standard threads on 303 Stainless . . . with thread extending to the head . . . and the overall job calling for an extremely fine finish. The best of the competitive oils used yielded only 600 pieces before serration and burring appeared, necessitating regrinding the chaser. Tycol Angrove 40 produced some 4,500 pieces with no defects apparent under magnification . . . and with no regrinding of the chaser throughout the entire run! If you can't top this – contact your local Tide Water Associated office for further information!

Over 300 Tycol industrial lubricants are at your disposal . . . engineered to fit the job!

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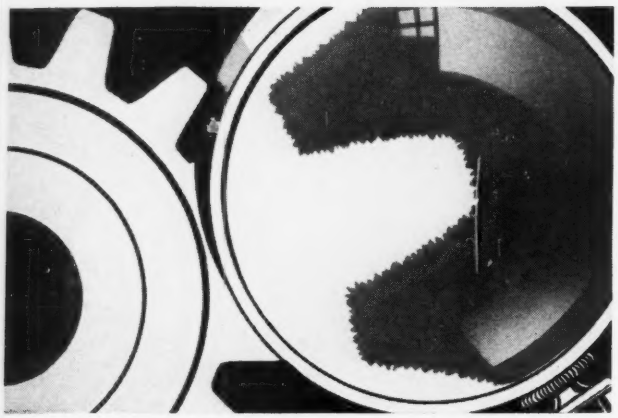
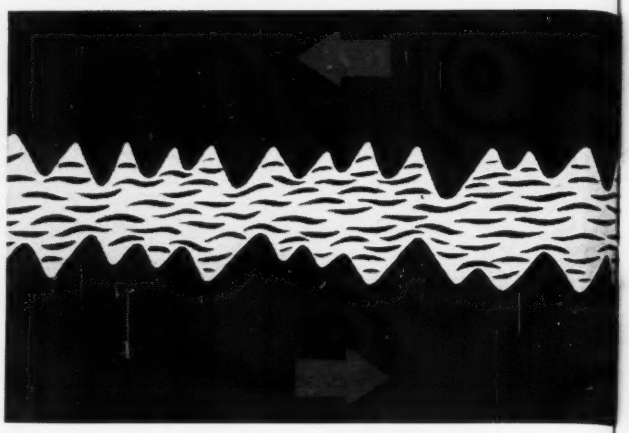
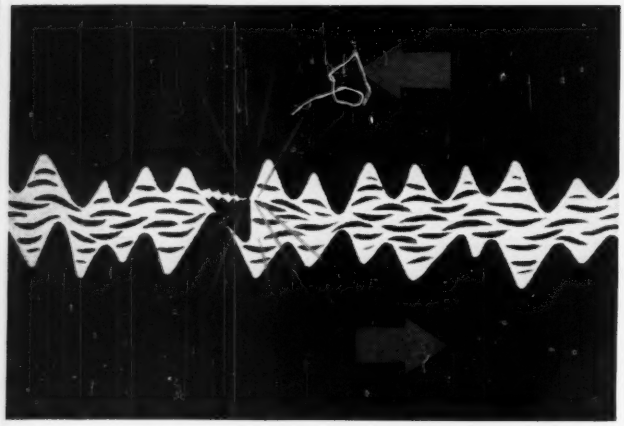
HOW SUNEP PREVENTS GEAR WEAR



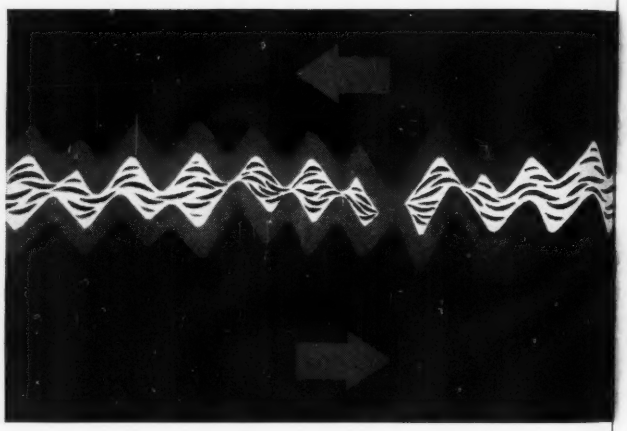
The surface of a gear may seem smooth, but when greatly magnified it looks something like this. The lubricant must keep these jagged edges apart.



With ordinary oil, a thick film is needed to keep these metal surfaces apart. The thickness of the film depends on the viscosity of the oil and other factors.



It is difficult to maintain a thick oil-film in everyday practice. The oil film thins out and semiboundary lubrication results. With ordinary oil, metal to metal contact occurs and gears wear.



Sunep coats the surfaces with a chemically bonded film which is virtually unbreakable. This film, although only about 0.0000001" thick, keeps the surfaces apart, withstands heavy loads and sudden shocks, prevents wear.

Sunep has many advantages. In addition to its outstanding load-carrying ability, it is noncorrosive; it keeps gears and bearings clean; it prevents rusting; it stays put. And its additives do not drop out during use or after prolonged storage. For more information about Sunep and how it protects your equipment, call your nearest Sun office or write SUN OIL COMPANY, Philadelphia 3, Pa., Dept. LE-10

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Palmoshield looks, feels and acts like palm oil; requires absolutely no changes in mill operation. Already most of the major tin plate mills in this country and Canada are using Palmoshield for regular mill operations or for production test runs.

. . . and grows and grows and grows!

Production records set with Palmoshield: Recently a leading steel company, over a run of 20 consecutive shifts, increased production 15% over the average output with imported oil. No additional investment, no increase in manpower, and no changes in mill operation. Yet 115 tons of steel were rolled for every 110 before. One company broke the single-shift record, and still another broke both 8-hour and 24-hour records.

Backed by 60 years of specialization: Palmoshield is the result of Ironsides' sixty years of specialized research in the field of scientific lubrication. For your own lubricating problems, call or write The Ironsides Company, 270 West Mound Street, Columbus, Ohio.

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cut pump-down time in half!
increase operating efficiency!
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Vacuum pumps

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WATER-OIL SEPARATOR

● **Removes moisture from vacuum pump oils; cuts pump-down time in half...** Keeps oil clean and dry. Enables pump to provide a high vacuum *without oil change* and eliminates pump "drying out" maintenance!

● **Eliminates changing or discarding of oil; allows maximum pump operating efficiency...** Increases production due to less downtime. Actually provides *2-way protection*—filtering out contaminants from oil as it separates water!

DRI-PURE is easy to install—can be hooked up to service one or more vacuum pumps depending on the application. *Quickly pays for itself in oil savings alone!*

WRITE TODAY for complete information. Equipment Bulletin available on request.

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MORGOIL

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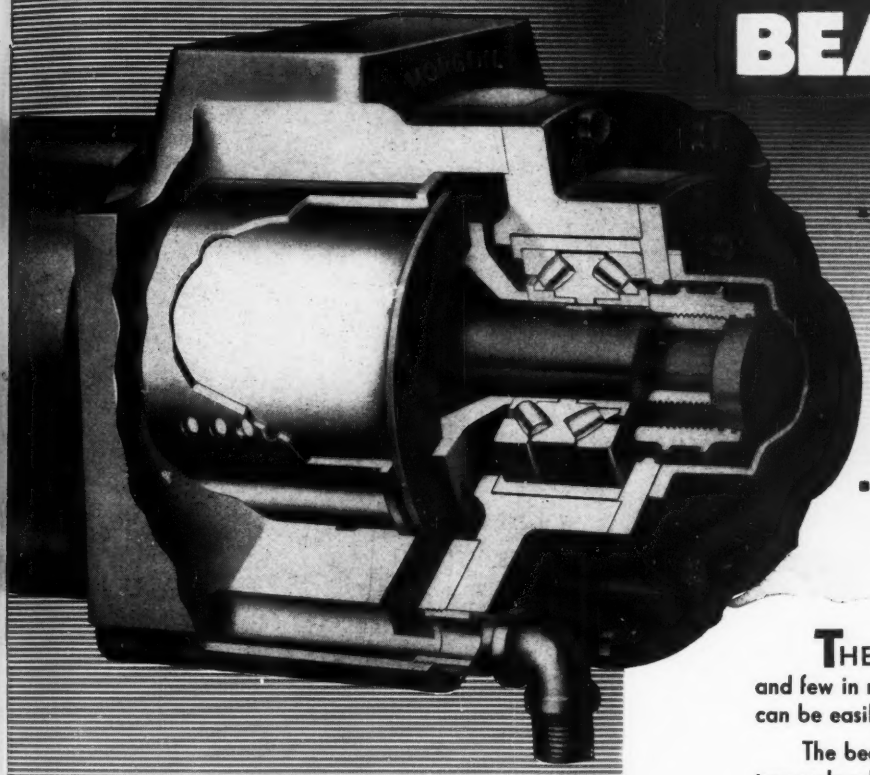

....insure accurate products

...last a long time

....require little maintenance

THE essential parts of the bearing are simple and few in number . . . making a compact unit that can be easily removed from the roll neck.

The bearing operates on a film of oil which has tremendous load carrying capacity, hence there is no wear. The load is distributed over a large area with no points of concentration, so fatigue failures of bearing parts are unknown.

The operation of a bearing under hydro-dynamic lubrication results in an extremely low coefficient of friction and low power requirements.

The absence of wear in the radial bearing and the close control of axial roll movement afforded by the thrust bearing, permits the rolling of both sheets and sectional products within extremely close tolerances.

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Gives white hot steel the "kid glove" treatment

GIGANTIC MACHINES with thousands of moving parts are used to shape white hot ingots into plates and sheets. All of these hard-working machine parts are subject to intense heat . . . normal in steel manufacturing. Because of the intense heat, lubrication is a serious problem.

U. S. Steel has tried various types of greases in order to eliminate the problem of oil burn-out. Now they're using a product of Shell Research, Shell Alvania Grease. Result: better lubricating action *at once*. Months after the original installations, rollers were removed and an excellent film of grease was still present.

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SHELL ALVANIA GREASE

- **resists oxidation**
- **will not squeeze out**
- **extends periods between overhauls**
- **provides exceptional lubrication in high or low operating temperatures.**

Shell Alvania Grease can cut down costly maintenance and save time and money in *your* plant. Write for technical information.

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telling the story of 'dag' dispersions

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Try *dry-film* lubrication . . . try **'dag'** Colloidal Graphite.

This unique material is not ordinary powdered graphite. It is high-purity, electric-furnace graphite, specially treated by Acheson Colloids to produce particles many, many times finer than those of graphite powder. Dispersed in a variety of liquid carriers — water, oils, volatile hydrocarbons, resin-solvent combinations—**'dag'** Colloidal Graphite goes where it should, and gets down to work.

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Try **'dag'** Colloidal Graphite for conveyor lubrication, for bearings, for moving parts subject to degreasing action, for anything that must move at high temperatures.

Details are available in Bulletins No. 424-12K and No. 435-12K.

Dispersions of molybdenum disulfide are available in various carriers. We are also equipped to do custom dispersing of solids in a wide variety of vehicles.

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try 'dag' resin-bonded dry films for permanent lubrication

ANNUAL MEETING HI-LITES

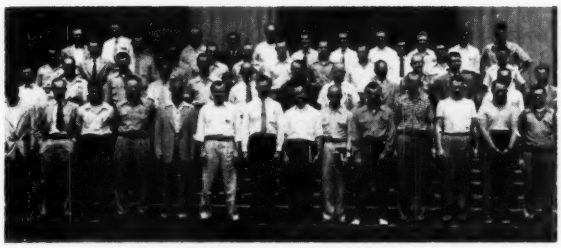
1954 ASLE ANNUAL MEETING & EXHIBIT
NETHERLANDS PLAZA HOTEL, CINCINNATI, APRIL 5-6-7

A partial list of papers to be presented at the 9th ASLE Annual Meeting & Exhibit include:

- Bearing Design Today
- Design & Application Of Oil Seals
- Effect Of Atmosphere On Friction & Wear
- Evaluation Of Solid Lubricants
- Fundamentals, by E. R. Booser
- Gear Lubrication
- Grease Rheology
- Hydraulics Today In Machine Tool & Coal Industry
- Industrial Plant Lubricant Disposal, by D. M. Cleaveland
- Lubricants & Cutting Oils
- Lubrication Application Problems & Their Solution In The Coal Industry (Panel)
- Lubrication Application Problems & Their Solution In The Steel Industry (Panel)
- Management (Cadillac Plant), by W. P. Youngclaus, Jr.
- Management (Hoover Co.), by A. F. Brewer
- Mechanism Of Rust Preventives
- Reaction Between Metals & Absorbed Films
- Reclamation Of Industrial Petroleum Products (Symposium)
- Relation Of Oil Properties To Molecular Structure
- Rust Prevention, by M. L. Langworthy
- Safety, by A. F. Brewer
- Synthetic Lubricants, by M. L. Langworthy
- Tests & Specifications (Symposium)

J. J. O'Connor, Chairman

Lube Lines



M.I.T. LUBRICATION ENGINEERING SHORT COURSE HI-LITES: Forty-eight students attended the ASLE sponsored Lubrication Engineering Short Course at the Massachusetts Institute of Technology in Cambridge, Massachusetts, June 16-26, 1953. The course was under the supervision of Prof. B. G. Rightmire, who is in charge of the Lubrication Laboratory in the M.I.T. Mechanical Engineering Department.

Pictured above, from left to right, are (1st row): R. A. Schleicher, J. E. Howard, R. K. Meyers, W. Kruck, E. E. McDole, W. C. Steen, E. D. Watson, W. C. Cook, R. W. Meals, L. J. Tirpak, N. A. Bracht, J. W. Hannigan, A. C. Brownell, P. Skerchock. (2nd row): F. Heymann, W. H. Fisher, T. D. Richards, L. D. New, H. E. Sipple, I. W. Paine, C. I. R. McDougall, J. LeMaitre, P. T. Green, R. O. Lovendahl, W. H. Voorheis. (3rd row): J. A. Box, B. G. Rightmire, D. G. Faust, C. G. Thompson, E. C. Lang, D. E. Benjamin, R. H. McCreery, E. Yatsko, R. E. Cassel, P. B. Burgess, C. M. Michaels, H. F. Eichhorn. (4th row): E. M. Kipp, F. L. Reynolds, C. A. Phalen, B. B. Wanderman, J. Y. Chou, D. A. Cardwell, L. Keto, R. S. Tooker, R. N. Fracalossi, J. A. Levene, W. Cassanos, R. C. Sheard.

WHO READS LUBRICATION ENGINEERING? Circulation of LE, as based on a survey completed August 14, and on a check of ASLE records August 17, 1953, revealed the following readers: Machine Maintenance 2336, Sales & Technical Sales 1390, Research & Development 955, Education 597, Machine Design 590, Other (Management, etc.) 306 (TOTAL 6174).

Industries covered, included: Petroleum 1508, Metalworking 1391, Education 520, Steel 514, Chemicals 318, Non-Ferrous Metals 317, Lubrication Equipment 301, Electrical Equipment 265, Government 249, Processing 170, Railroads 161, Textile 117, Mining 81, Food 70, Advertising Agencies 51, Aircraft 44, Glass 43, Paper 33, Marine 21 (TOTAL 6174).

NEW METAL-WORKING LUBRICANT SCHOOL: A metal-working lubricant school to provide instruction in wax lubricants has been established by S. C. Johnson & Son, Inc., makers of wax products, at the company's Racine, Wisconsin, headquarters. Created to train salesmen of Johnson's distributors in the fundamentals of wax chemistry and in the use of Johnson's new wax cutting fluids and lubricants, the curriculum consists of lectures and demonstrations in the properties of wax and its use in metal forming, cutting and machining, piercing and blanking; proper methods of cleaning are covered, and demonstrations made on an instrumented test press, a surface grinder, a tool grinder, and a drill press. It is intended that this become a permanent school not only to instruct the salesmen of Johnson's distributors, but for customers who want to learn more about the properties of wax lubricants.

ASTM ACCEPTS 63 NEW MATERIALS SPECIFICATIONS: At the 1953 Annual Meeting of the American Society for Testing Materials, 63 new specifications and tests were approved. Included were: Methods of: Test for effect of grease on copper (D1261-53T), Test for lead in new and used greases (D1262-53T), Test for leakage tendencies of automotive wheel bearing grease (D1263-53T), Test for water washout characteristics of lubricating greases (D1264-53T), Sampling liquefied petroleum gas (D1265-53T), Test for sulfur in petroleum products and liquefied petroleum gases by the Co_2-O_2 lamp method (D1266-53T), Test for vapor pressure of liquefied petroleum gas (D1267-53T), Test for unsaturated light hydrocarbons (silver—mercuric nitrate absorption) (D1268-53T), and test for polarographic determination of tetraethyllead in gasoline (D1269-53T). All will be published later in the 1953 Supplement to the Book of ASTM Standards; individual copies of specifications will be available at a later date from ASTM Headquarters, 1916 Race St., Phila. 3, Pa., at a nominal charge, usually 25 cents each, but slightly higher on standards over 16 pages.

CORROSION ENGINEERS TO HOLD REGIONAL CONFERENCE: Corrosion problems affecting pipelines, oil refineries and aircraft will be the principal topics of discussion at the 3rd Annual Conference of the National Association of Corrosion Engineers, Western Region, November 19 & 20, Biltmore Hotel, Los Angeles. Technical papers on gas and water pipeline corrosion will be presented during the afternoon session of Nov. 19; papers on corrosion as it affects the aircraft industry will be presented during the morning session of Nov. 20, with problems of petroleum refineries covered during the afternoon session. The conference ties in with a three-day course in cathodic protection to be offered by the Univ. of Calif. at Los Angeles on Nov. 16, 17 & 18.

SAND IN THE GEARS (In part, from Industrial Bulletin of A. D. Little, Inc., July '53, No. 300): Since the invention of the wheel, man has been plagued by the problem of lubricating the bearing on which it rotates. Today, even the usual automobile service station must stock several kinds of grease to meet the variety of destructive conditions found in the machine. Recently, however, new greases thickened with solid materials rather than with the standard soaps have appeared; they stand up well enough to carry the industry closer to the ideal of the multi-purpose lubricants.

Some of the new solid-type thickeners for petroleum lube oils bear a startling resemblance to ordinary dirt, the last thing one would normally put into a bearing. They are, however, composed of such fine particles and are so highly refined to rid them of abrasive materials that they do not contribute to bearing wear. Their chief value lies in

(Continued on page 268)

LUBRICATION REQUIREMENTS OF GEARS AS SEEN BY A GEAR ENGINEER*

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INTRODUCTION: The exact mechanics of gear lubrication are not clearly understood at the present time. In the case of journal bearings where the unit loading is relatively low compared to the unit loading of gears, lubrication difficulties may be experienced. For example, in the case of journal bearings it is not uncommon to experience difficulties when the unit loading is in the neighborhood of 2000 to 3000 pounds per square inch, whereas, in the case of a gear tooth, a conservative loading may be in the neighborhood of 60,000 to 70,000 pounds per square inch. Gear teeth have been and are being successfully lubricated at tooth pressures in excess of 180,000 pounds per square inch. Lubrication studies are being conducted and a number of theories have been advanced to explain how the lubricating film is established and why it is able to withstand such extremely high surface loads.

Some of the factors which could affect the lubrication of a gear train and which might lead one to conclude that the lubricant was at fault are: (1) Accuracy of tooth spacing, (2) Tooth surface finish, (3) Tooth modification, (4) Sliding velocity.

ACCURACY OF TOOTH SPACING: If the teeth on a high speed gear are not accurately spaced the tooth surfaces are subjected to dynamic loads during operation and the severity of the dynamic loads will limit the useful power which can be transmitted successfully. For example, consider a high speed train of gears in which the moments of inertia of the prime mover and the load are high. If sizable errors in tooth spacing exist, instantaneous changes in angular velocity will result. When we consider a tooth which is only in contact for one ten thousandths part of a second it becomes apparent that if this tooth is improperly spaced, the blow which will be transmitted from one tooth to another will pierce the oil film and may also physically damage the two teeth.

Many design specifications indicate that as the speed is increased, the useful power which may be transmitted will decrease. Where such specifications are used, therefore, the theoretical load which the gears should carry is calculated and then the actual power which the gears will carry is determined by multiplying this theoretical valve by some factor called the "velocity factor."

Where the teeth are very accurately spaced it has been our experience that it is not necessary to use velocity factors and tests have proved to us that as the speed is increased, the load which may be carried can also be increased and that it is not necessary to decrease the theoretical power by a velocity factor. By accurately spaced teeth we mean teeth which have an accuracy of tooth to tooth spacing such that the tooth to tooth error does not exceed 0.0002".

TOOTH SURFACE FINISH: The surface finish of a gear tooth is of extreme importance and the surface finish which may be considered satisfactory in one application may be entirely unsatisfactory in another application. If the tooth surface finish is rough, it is then composed of little hills and valleys and the mating surfaces may be likened to two files in contact. If lubricant is applied to these surfaces, these little hills may pierce the lubricating film and allow the hills of the two mating surfaces to come in contact, or the hills on one surface may fall into the valleys on the other surface.

Under this latter condition there is an interference between the two surfaces and in order for sliding to take place the hills must be removed either by shearing off or being bent over by the mating imperfection.

If the viscosity of the lubricant and the load applied between the two surfaces is such that the force applied is not sufficient to pierce the oil film, the teeth will ride on the lubricant and metal to metal contact will not be experienced. As the load is increased, however, a point will be reached at which time the hills on one tooth will contact the hills in the other, or the hills on one tooth will fall into the valley of another and the mechanical shearing of the metal or the bending over of the hill will result in heat being generated.

We have conducted tests to demonstrate this phenomenon and one such test will be briefly described here. A set of spur gears of approximately 2½ diametral pitch, 20 teeth on the pinion and 75 teeth on the gear and with a surface finish of approximately 100 microinches was installed in a gear box and provided with forced lubrication using a standard 2190 T lubricant. The unit was operated at rated speed and the load increased to 100,000 p.s.i. immediately. The casing became filled with a dense smoke indicating that heating on the

*Sponsored by the ASLE Technical Committees on Lubrication Fundamentals, and Physical Properties of Lubricants, and presented at the ASLE 8th Annual Meeting, Boston, April 13, 1953.

surface of the tooth raised this surface to a temperature sufficient to cause smoking of the oil. This unit was operated continuously under this high load and as time passed, the amount of smoke decreased until finally the unit was operating without any smoke. During the period when smoking was taking place, the oil was being consumed and had to be replaced by adding oil to the sump tank. After operating for several hours the tooth surface was again measured and found to be in the neighborhood of twenty microinches. There were a number of pits on the surface of course due to locallized stress concentration but in between the pits, the tooth surfaces had been polished.

It has often been stated that if the tooth surface is too highly refined, the surfaces will not sustain a lubricating oil film. This statement should be qualified because the speed at which the two surfaces slide with respect to each other has a great deal to do with the ability of lubricants to separate the two surfaces. For example, if a tooth surface is very highly polished and if the sliding velocity is extremely low, it is entirely possible that the hydrodynamic film will not be established and the conclusion drawn is that the tooth surface finish is too finely finished for the particular application. On the other hand, where the sliding velocity is extremely high, as in the case of gears operating with a pitch line velocity in the neighborhood of 30 to 50 thousand feet per minute an extremely fine surface finish is desirable and a hydrodynamic film is established and the teeth can be satisfactorily lubricated. Under these high velocities in the interest of obtaining efficiency it is desirable to have a tooth surface finish as fine as possible.

It has also often been said that if the surface was 0 microinches or absolutely perfect, no lubricant would be required. This is true as long as the load between the two surfaces is kept at 0 or in the very low value. As the load is increased, however, the surfaces deflect and the plastic working of these surfaces results in heat being generated which must be conducted away by a lubricant. To date no one has produced a tooth the surface finish of which is so perfect that no lubricant is necessary.

TOOTH MODIFICATION: The teeth on a gear may be perfectly spaced when checked with instruments used to determine the accuracy of the tooth spacing and yet difficulty may be encountered during operation. If we consider two gears in mesh the tooth spacing of which are perfect, at no load smooth engagement will result. If the load is applied and this load is steadily increased, teeth which are sharing the load begin to deflect which has the effect of displacing these teeth and since the teeth in mesh are displaced, those which are entering engagement will also be displaced. What may actually happen is that the corner of a tooth strikes the mating tooth in such a manner that the oil film is scraped off. This can be eliminated by tooth modification. This modification consists of increasing the radius of curvature at the tooth tip a certain amount and down a certain depth and is in effect similar to the effect of the upturned end of a pair of water skis when pulled through the water at a relatively high speed. Gears which have experienced a great deal of difficulty when operating at high velocity at heavy loads have been thus modified with the result that operation has been entirely satisfactory from all standpoints.

SLIDING VELOCITY: Two teeth in engagement are subject to sliding between the two surfaces at all times during the engagement except at the instant when rolling takes place on the pitch line. The percentage of sliding, or the sliding velocity between the two surfaces is a percentage of the pitch line velocity. The maximum sliding velocity is a function of the tooth proportions. If extremely long addendum teeth are used then the sliding velocity at the tip of the tooth is high and unless the teeth are properly modified and the surface finish highly refined, burning of the tips of the teeth may result. Extremely long addendum teeth have been placed on test and operated until the tips of the teeth turned blue indicating excessive heating in this particular region. The importance of tooth proportions, that is the length of the tooth and the sliding velocity should be considered wherever difficulties of this type are being experienced. Thus far we have considered four factors, namely, the accuracy of tooth spacing, surface finish, tooth modification, and sliding velocity. These factors are therefore, dependent upon the manufacturing processes to obtain accuracy of tooth spacing and surface finish and tooth design to provide the proper tooth modifications and limit sliding velocity to values which experience has indicated are reasonable.

It is quite natural that if a gear designer examines one of his units which has been in service for some time and finds that this unit is showing distress, his first reaction is usually to blame the lubricant. Some of the most common types of distress appearing on gear teeth are pitting, abrasive wear, scoring, burning and rolling.

PITTING: There are a number of terms used to describe pitting, for example, initial or corrected pitting, destructive pitting, etc. The surface of a tooth which has been pitted appears to have particles of material plucked from the surface thus leaving craters on the surface. Since pitting may occur on teeth of units which are not loaded excessively, it is safe to assume that slight surface imperfections resulted in locallized points of high pressure. For example, if a gear tooth is finished in such a manner that machine marks are present on the surface, the resulting high points result in load concentrations at locallized points which result in pits being formed because these particular areas are overloaded. In general, if the speed and load are held constant, pitting will cease after a period of time and the areas in between the pits will become polished and no further pits will result. This phenomena has been described in numerous publications and is common in the gear industry on gears the surface finish of which is not too highly refined.

ABRASIVE WEAR: Abrasive wear is in general caused by foreign material in the lubricating oil and has the appearance of a tooth which has been rubbed with emery cloth or sandpaper. Since the direction of sliding is radial, the abraided surface will show scratches at right angles to the axis of the tooth.

SCORING: Scoring is in general the result of small particles having become welded to the mating elements and producing scratches extending radially along the flank of the tooth. This could be caused by the high point becoming so warm that it adhered to the mating element and produced the scratch because it is trapped in between the two loaded surfaces which are moving in opposite directions

BURNING: Burning results in a discoloration of the tooth surfaces referred to previously in discussing the long addendum tooth and indicates that for some reason or another heat was generated between the two surfaces and was not carried away by the lubricating oil, or that the load was too high or that the coefficient of friction between the two surfaces was excessive.

ROLLING: It has been noted that some teeth under load will form a little burr along the tip of the tooth which indicates that the metal is being displaced from the surface. It appears that the heat was conducted away because the teeth are not discolored indicating excessive heating. Rolling seems to be due to a plastic flow of the material and has been observed on teeth the loading of which for all intents and purposes could be considered moderate.

Pitting, abrasive wear, and scoring and burning might be considered as lubrication failures. If pitting is encountered and since pitting indicates points of locallized stress concentration, it is highly improbable that a lubricant could be applied which would prevent pitting. Assume for instance that a tooth surface was so designed that if all the load were distributed across the face, the surface would be subjected to a stress of 90,000 lbs. per square inch. If, however, a number of bumps were present on the tooth surfaces and the total area of these bumps was equal to ten per cent of the total tooth area, then the unit pressure would be equal to 900,000 lbs. per square inch on the point thus loaded and this excessive loading would result in pitting.

If sufficient lubricant is present to keep the temperature of this particular point below the temperature at which it would adhere to the mating member, the particle which pitted out would go into the lubricating oil and be picked up by the magnetic strainer. If, on the other hand, the temperature did increase to a point at which seizure took place, the material would adhere to the mating member and a score mark would be produced on the tooth surface. If a tooth shows burring or discoloration this indicates that the heat has not been carried away and that the surface of the tooth has been heated up but not necessarily to a point where seizure will take place.

If abrasion is noted, this indicates that the lubricating system should be cleaned and the lubricant filtered to remove the foreign material from the system.

Let us now imagine that we happen to be a lubrication engineer and that on given day we are called upon to examine the following: (1) A marine gear which has become noisy and is now indicating wear by the fact that metal is being picked up in the magnetic strainer. (2) A marine gear that operated for ten years and during the last year has begun to show undercutting and metallic particles in the lubricating oil. (3) A new gear, heavily loaded which shows some very fine pits after having been in operation for only one week. (4) A gear unit driving an induced draft fan which has become noisy and yet the tooth surfaces appear to be in good condition.

In the case of the first unit we find that for some reason or another a piece of steel such as a nut or a bolt has gone through mesh. The teeth have been corrected by filing off the damaged portion of the tooth. Due to filing, the tooth surface has been roughened and wear started in this area. The small steel particles which appear due to this wear contaminate the oil and induce wear on other parts of the tooth surfaces.

The system should be flushed and the oil filtered. It may be desirable to put in a heavier grade of oil until the teeth have again obtained a polish.

In the second case, the gear operated for ten (10) years and then suddenly began to show wear. A careful examination will probably indicate dirty oil. The system should be cleaned and the oil filtered. If the wear is noted soon enough and the condition of the oil corrected the gear should polish up. If the wear has been allowed to go too long it may be desirable to use a heavier oil until the teeth again take on a polish after which oil of the original viscosity may be used.

In the third case, since the gear is a new one and the small pits are noted at an early stage in the operation of the unit, pitting indicates that there are slight bumps or rough spots on the tooth surface which result in stress concentration. Since this gear is heavily loaded the best solution is to use an oil which has anti-welding properties or an extreme pressure oil. Under the action of this oil the teeth should polish up.

In the case of the induced draft fan drive gear, it will probably be discovered that the pinion teeth while they look good have worn unevenly. That is to say, the pitch line is no longer concentric with the axis of rotation.

A careful examination may reveal that the load is not constant and the torque varies. The oil is contaminated with fly ash and the cyclic wear has taken place.

The solution is to recut the gear and pinion, clean the system and if possible, eliminate the load variation or at least reduce their magnitude so that no further harm will be done.

In closing it should be pointed out that we have observed numerous cases where a tooth has broken out under heavy loads, and that the lubricant has been stronger than the tooth. The tooth surface was highly polished, free from pits, abrasion, rolling or galling.

The problem of lubricating a gear is made a great deal easier if all the factors which go into making up an accurate gear properly designed as to tooth proportion and modifications are observed.

PLAN NOW to attend

1954 ASLE Annual Meeting & Exhibit

Netherlands Plaza Hotel, Cincinnati, April 5-6-7

THE SIGNIFICANCE OF GREASE TESTING*

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As our fundamental and practical knowledge of technology continually grows, it becomes increasingly important that we devise reliable short time tests which will permit us to distinguish among the array of new materials being submitted for consideration. This need exists in the field of lubricating greases as much as in any other. Unfortunately there is still no bench test that will completely answer all the questions so well as actual performance in the field or in a closely simulated operation. We can, however, often get important information about one or more factors in the performance of a grease in a short time with simple laboratory evaluations.

This discussion will indicate the significance of a number of grease tests which have been in use in our laboratory and elsewhere. In addition to standard bench tests, this review will also cover a number of other short time evaluations developed to simulate more closely actual equipment operation. By removing greases from complex operating devices for study under controlled laboratory conditions, these functional tests are frequently quite useful in establishing the fundamental properties of greases which are important to good performance. Some of these tests, for instance, measure properties such as consistency and shear stabiilty which are functions largely of the soap; others, like low temperature torque and volatility, depend more on the oil; and some, such as bleeding and oxidation, involve both.

In addition to covering tests designed to determine the performance characteristics of greases, a discussion is also given of simple means for checking the condition of used greases from the field.

SOAP STRUCTURE TESTS: The function of a soap is to provide a spongy structure which will serve as a storehouse for oil which then may bleed slowly into the moving parts requiring lubrication[1]. This soap structure is of a fibrous nature which can be seen through the use of the electron microscope. With the microscope, the length-to-diameter ratio of the fibrous crystallites of soap has been measured. The importance of this ratio has been demonstrated by experiments[2] in which nylon particles of known L/D ratio were suspended in oil, simulating a soap structure. The penetrations of the mixtures were similar to those of regular soap base greases having the same L/D ratio. The bleeding rates were, of course, much larger because the particles, and therefore the interfiber spacings, were much larger.

Also important is the phase stability of the soap structure, both chemical and physical. The chemical stability is often a function of temperature and sometimes of the presence of stabilizers, such as water in calcium greases[3]. In the absence of the stabilizer the structure breaks down, resulting in a fluid instead of a grease. The structure may also be destroyed by excessive working. With an ideal grease, the reformation of the fibrous network is sufficiently rapid so that ordinary working or churning will not cause the grease to become fluid.

The structure of a grease is a function also of the chemical nature of the soap. Again, the electron microscope has permitted detection of interesting differences between the various fatty acid chains which form soap structures. Lithium hydroxystearate, for instance, forms twisted chains while the normal stearate forms a straight fiber[4]. Different fatty acid soaps also vary in their ability to maintain structure under adverse working conditions or temperatures, with those of around fourteen to eighteen carbon atoms per chain giving best results.

CONSISTENCY TESTS: The quickest and simplest tests for evaluating soap structure characteristics are the micro[5] and macro[6] penetration measurements obtained by use of penetrometers. Both these penetrometers are arbitrarily designed cones which, when dropped from a specified height, will sink to a depth which is an index of the stiffness of the grease. The usual penetration value in decimillimeters has little significance in itself although it is related to the yield strength[7] which is the force necessary to cause distortion of the grease. A yield value of less than 981 dynes per square centimeter, for instance, means a one centimeter cube of grease will slump under its own weight. This yield value is equivalent to an ASTM penetration of about 400, too soft for most purposes. The average general purpose greases must be stiff enough to resist excessive slumping but still be able to flow into spaces where they are needed. These usually have a penetration of 260 to 290. There are applications where channeling is required to keep grease from falling into the path of the rolling elements and creating excessive friction and shear breakdown of the grease structure. A penetration of about 200 to 250 for such channeling greases is frequently preferable in double sealed and double shielded bearings. Much softer greases are necessary where more feedability is necessary in gears and in difficult to lubricate bearings. A 355 to 385 penetration has been found best for railway journal bearings[8].

STRUCTURE BREAKDOWN TESTS: Other tests which provide information about the structure of a grease include working tests and whipping tests. The ASTM working test[6] forces a grease through

*Sponsored by the ASLE Technical Committee on Lubrication Fundamentals and presented at the ASLE 8th Annual Meeting, Boston, April 15, 1953.

a perforated plate at a rate of 60 times per minute. Measurements of penetration are made before and after varying periods of working to determine how a grease might stand up in operation.

A whipping test using an over-packed ball bearing housing has been found a particularly effective measure of stability. Although somewhat similar, it differs from the ASTM worker in that the rate of working of a non-channeling grease is greatly increased, with no time for recovery of the structure between the shearings by the fast moving bearing parts. The test equipment is a regular No. 306 ball bearing with a lock nut in an 1800 rpm motor. The bearing is packed completely full of grease and run for 24 hours. Greases which are not able to channel, that is, to be pushed aside and remain pushed aside, are badly churned in the ball path and become fluid. That the information derived from this test is different from that obtained with the ASTM working test is shown in Table I where the original and final micropenetrations for several greases are tabulated along with their macro penetrations measured after ASTM working. It can be seen that some greases which withstand working cannot withstand whipping; and conversely, some which channel so well as to remain unaffected in a bearing whipping test are broken down under the continuous shearing of the ASTM worker and would appear to be unsatisfactory. The whipping test seems to provide better correlation with operational tests for large high-speed bearings and sealed and shielded ball bearings.

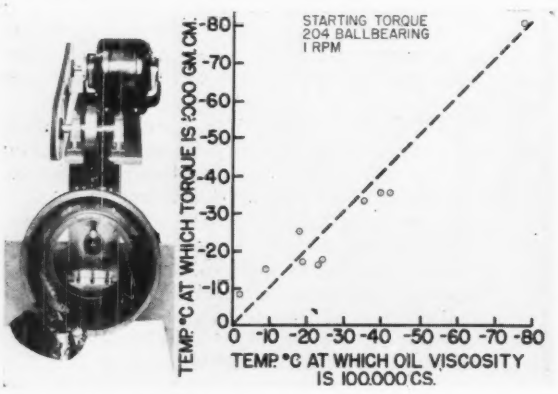
STARTING TORQUE
204 BALLBEARING
1 RPM
TEMP °C AT WHICH TORQUE IS 2000 GM CM.
-80 -70 -60 -50 -40 -30 -20 -10 0
0 -10 -20 -30 -40 -50 -60 -70 -80
TEMP °C AT WHICH OIL VISCOSITY IS 100,000 CS.

Fig. 1 (left) Slump Tester.

Fig. 2 (right) Relation of grease oil viscosity to limiting low temperature torque.

SLUMPING & CHANNELING TESTS: Additional properties of grease dependent on the soap structure are slumping and channeling. These are apparently functions of both structure and adhesiveness. A slumping tester developed here is shown in Fig. 1. It consists of a railway traction motor bearing cap which is 60% filled with grease and vibrated at 30 cycles per second with a 1/8" stroke at room temperature. The time is measured for the grease to touch a horizontal plate 1 inch below the bottom of the grease cavity. The data of Table II show that for a given grease the slumping time is inversely related to the penetration. When comparing different greases, however, this relationship does not necessarily hold. Therefore, penetration cannot be the sole criterion of a good channeling grease. The other factor is probably adhesiveness, that is, the ability of a grease to adhere to a metal. Even an otherwise good channeling grease will not maintain its position if, while remaining rigid, it falls away from the walls of the bearing housing.

The ability to channel and resist slumping may also be observed by continuously recording torque or temperature in a bearing. Chunks of grease falling into the path of a fast moving set of rolling elements will cause sudden rises in these parameters. This is particularly objectionable in large railway traction motor bearings where the magnitude of the temperature rise can be serious because of the large volume of slumping grease.

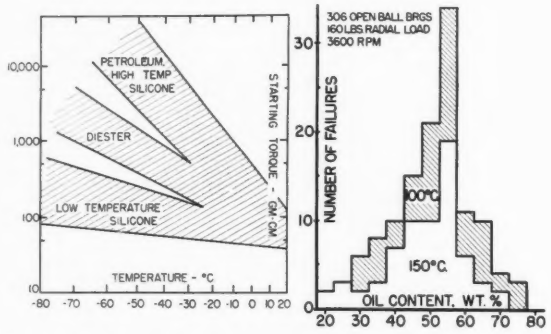

Fig. 3 (left) Low temperature grease torque groupings.

Fig. 4 (right) Oil content of petroleum grease at failure.

BLEEDING TESTS: Bleeding is also dependent largely on the soap structure. There are two mechanisms responsible for the loss of fluid from a grease: one, the hydrostatic pressure forcing the oil out of the interstitial spaces and the other, a squeezing out by a shrinking soap structure. The net result in either case can usually be expressed by the following equation [9, 10]: $B = t/(a + bt)$, where "B" is the oil loss in per cent of the original grease weight at time "t" in hours for any given temperature. Then $1/a$ is the original rate of bleeding and $1/b$ is the final total loss. Best greases will have a large "a", that is, a low initial bleeding rate, and a small "b" representing a high oil utilization and a long useful life. Low bleeding rates are usually satisfactory because very small amounts of oil are required to keep a bearing running. A standard No. 306 ball bearing, for instance, will run at 3600 rpm under 160 lb. radial load for 1000 hrs. with half a drop of oil[11]. As shown later, ball bearing lubrication failures start to occur when the grease has lost only 60% of its oil, so maximum efficiency in bleeding rate is desirable and should be determined before application.

A common and simple method of measuring this bleeding rate is described in the MIL-G-3278 specification. It consists of placing ten grams of grease in a wire mesh cone hung in a closed vessel which catches all the oil bleeding from the grease. The vessel plus oil is weighed after 30 hrs at 100 C., and the bleeding calculated as weight per cent of oil.

The data of Table III were obtained using a modification of this test. The constants "a" and "b" were calculated from the bleeding rates and tabulated. Also in Table III are the calculated bleeding rates at the failure time in life tests on 306 bearings running at 3600 rpm at the stated temperatures. It can be seen that the bleeding rate required is higher as the temperature increases. This may be due to several factors including increased volatility, greater mobility, and increased

oxidation rate, all of which result in faster loss of oil from the bearing races. Although it may be fortuitous, the bleeding rates at failure are of the same order of magnitude whenever the viscosities of the petroleum oils are the same.

APPARENT VISCOSITY: The apparent viscosity of a grease is a seldom measured parameter which is fundamentally important. It is the ability of a grease to become fluid at high rates of shear, and to revert to a relatively rigid structure when shearing stops, that permits it both to lubricate well and to stay where it is needed. The apparent viscosity has practical importance for designers of pumping systems and automatic feed systems. At low shear rates, grease viscosity is largely a function of soap structure except at very low temperatures where the viscosity of the oil becomes the more important factor as discussed later under low temperature torque.

OIL PHASE TESTS: Petroleum oils are classified as either naphthenic, aromatic or paraffinic. The paraffinic oils generally are more stable towards oxidation, have lower specific gravities and higher viscosity indices. The naphthenic and aromatic oils generally permit easier grease manufacture, but for low temperature greases they have a disadvantage in that their viscosities increase more rapidly as the temperature decreases.

VOLATILITY: Low volatility of the oil component at the operating temperature is a desirable grease characteristic. This property may be studied by procedures described in the ASTM D972-51T test and by a modification of the AN-G-5a bleeding test. These are designed primarily to measure the weight loss at periodic intervals at any given temperature. The higher the temperature for which a grease is designed, the lower must be the volatility of the oil. In general, this means use of higher viscosity oils. From high temperature ball bearing tests with several diester and polyester greases, the limiting vapor pressure of the oil for 1000 hrs. of operation seems to be about 0.03 to 0.06 mm. of mercury as indicated in Table IV. At very high temperatures, however, the rate of oxidation of petroleum greases frequently becomes the limiting factor.

VISCOSITY: If oil viscosity in a grease is made too high, losses at operating temperatures may be increased and the torque required to start a packed ball or roller bearing may become excessive at low temperatures. The latter property can be tested readily with an apparatus which measures the torque necessary to start and turn a grease packed ball bearing at a constant speed of 1 rpm. The temperature at which the starting torque in a No. 204 ball bearing becomes 1000 gramcentimeters is the lower limit for many applications where moderate torque is available. These values and the temperatures at which the viscosities of the oils in the greases become 100,000 centi-stokes are shown in Table V and Fig. 2. The correlation indicates that the soap is a neglible factor compared to the oil viscosity at sub-zero temperatures. The plots of log starting torque vs. reciprocal temperature for a wide variety of greases in 204 bearings are combined in Fig. 3. It can be seen that the greases are divided into three general categories: (1) petroleum and high temperature silicone, (2) diester, and (3) silicone-diester and low temperature silicone.

LUBRICITY: The principal application of grease lies in the lubrication of rolling element bearings. These generally do not require too much lubricity in the grease except for high load and high speed operation. With silicone formulations, the limiting speed factor (bearing bore in mm. times the RPM shaft speed) is approximately 150,000 to 200,000 and high load operation is unsatisfactory.

TABLE I		ASTM Worked Penetration (6)					Micropenetration (5)	
Soap	Oil	Strokes	60	10^3	10^4	10^5	10^6	Orig.	Whipped*
Na-Ca	Paraffin blend		280	315	335	340	364	75	>420
"	" "		300	319	338	357	365	73	385
"	" "		298	302	312	325	324	101	168
Na	Naphthenic		285	300	345	364	>400	85	85
Li	Naphthenic		230	-	246	262	-	57	63
"	"		275	-	299	295	322	90	>420
"	Diester		273	283	297	298	-	61	110

*After 24 hours operation in 1800 rpm overpacked 306 bearing housing at room temperature.

TABLE II	Oil Viscosity		Slumping Time
Soap	SUS, 100°F.	ASTM Worked Penetration	at 77°F, Minutes
Na-Ca	300	230	>30
Na	500	200	>30
Sr	2200	290	26
Na	245	290	10
Na-Ca	300	280	0.8
Li	500	280	0.5
	Single Grease Worked to Different Penetrations		
Na	300	300	0.1
"	"	297	0.1
"	"	256	1.3
"	"	233	13

TABLE III			a	b	Bleeding Rate at Failure,*	Oil Viscosity,
Soap	Oil	Temp °C	Hours/Percent	Percent^{-1}	Percent/Hour	Centistks.
Na-Ca	Petroleum	100	5	0.134	1.2×10^{-3}	16.5
		125	27.9	0.018	6.7 "	9.5
		150	6.5	0.017	19.1 "	6.0
Sr	Petroleum	44	57	0.063		
		75	57	0.041		
		100	19	0.34	0.14 "	32.0
		125	5	0.025	0.9 "	16.5
		150	2	0.023	4.5 "	9.5
Li	Diester	75	37	0.253		
		100	20	0.111	0.37 "	13.5
		125	5.5	0.066		
		150	2	0.076	1.2 "	1.8
Li	Silicone	44	350	0.462		
		75	50	0.189		
		100	15	0.113		
		125	15	0.105		
		150	12	0.088	0.02 "	7.6

*Calculated from "a" and "b" for failure time in 3600 rpm; No. 306 open ball bearing grease life tests.

Table I (top) Penetration changes after working and whipping.

Table II (middle) Slumping time of various petroleum base greases.

Table III (bottom) Bleeding rate constants.

Many other devices such as cams, pivots, ways, lead screws and many gears require good lubricity. A common and simple device for measuring wear under severe sliding conditions is the Navy Gear Wear Tester, described in MIL-G-3278 specification, in which a stainless steel and a brass spiral gear set are packed with grease and run an arbitrary number of reciprocating motion cycles under load. The gears are weighed before and after the run, and the loss of weight is the measure of wear. A wide range in the wear characteristics of petroleum greases is observed in the wear tester. Soft, high bleeding greases generally give low wear while the stiffer and drier lubricants develop early failure. Although stainless steel and brass is the standard, other combinations may be and should be used if radically different greases such as silicones are being tested, because lubricity will vary with the metal combination[12].

OXIDATION & CORROSION TESTS: There are various chemical factors involved in the operation of grease. The principal one is oxidation. Oxidation of the oil results in gums which solidify on cooling and volatile products which result in loss of fluid. The oxidation of the soap usually results in loss of the network structure. In steel ball bearings, however, the bearing will run so long as any fluid is present, because the steel is not attacked by the oxidation products.

The standard test which measures the oxidation resistance of greases is the Norma-Hoffman test in

which the rate of absorption of oxygen by the grease in a bomb is observed. The results of this test are related to the shelf life of grease as shown in Table VI.

More recently, a dynamic tester has been developed in which a grease packed ball bearing is rotated in a similar oxygen bomb. This tester has the advantage of simulating more closely the applications of greases.

The second major source of chemical trouble is corrosion, particularly rusting of steel. The usual sodium soaps in grease provide excellent resistance to corrosion in the presence of small amounts of water. They do this in two ways: first, they absorb small amounts of water and so keep it away from the metal surface; second, when even larger amounts of water get into the bearing, the soda soaps will dissolve in the water and by being adsorbed on the metal surface, protect it from corrosion. If large enough quantities of water are involved, the soluble sodium soaps may be washed entirely away and the metal left unprotected.

Protection against the grease being completely washed away is provided by using grease made of water insoluble thickeners such as strontium, calcium, barium and lithium soaps. These water insoluble soaps do not act so well as soda soaps as inhibitors of rust, but they do resist water washing. By addition of rust inhibitors, this type of grease may be provided with chemical rust inhibition as well as resistance to water washing. There are various tests to measure water resistance. One consisting of spraying water on a running bearing is described in the AN-G-5a specification. A humidity cabinet test using greased bearings is outlined in the MIL-L-4343 specification.

TABLE IV

Grease Composition		High Temperature Limit for 1000 Hrs. 306 Bearing Life	
Oil	Soap	Temp. °C*	Oil Vapor Pressure, mm. Hg**
Iso-amyl Sebacate	Li	110	0.05
Diethylhexyl Sebacate	Li	130	0.03
Diethylhexyl Azelate	Li	150	0.06
Polyester	Li	150	0.03

*Using a life factor of 1.5 per 10°C. to translate life data obtained at 100°C. and 150°C.

**Estimated from Standard Oil Development Company data.

TABLE V

Soap Type	Oil Type	Oil 100°F. Visc., Cs.	Viscosity Index	Low Temperature Limit, °F. 1000 Gm-Cm Torque in No. 204 Bearing	100,000 Centistoke Oil Visc.
Sr	Petroleum	475	106	9	1
Na-Ca	Petroleum	167	67	-15	-9
Na	Petroleum	110	90	-17	-23
Li	Petroleum	108	74	-18	-19
Na	Petroleum	67	63	-18	-24
Li	Petroleum	108	66	-26	-17
Na-Ca	Petroleum	65	85	-33	-35
Na-Ca	Petroleum	38	65	-35	-40
Li	High Temp. Silicone	95	152	-35	-43
Li	Diester	13	154	-80	-79
SiO_2	Low Temp. Silicone	70	157	<-100	<-100

a. Values extrapolated on ASTM Viscosity-Temperature Chart.

TABLE VI

Induction Period (hrs.)	Shelf Life (yrs.)
50	1
150	2-3
500	4

Table IV (top) Influence of vapor pressure on grease performance at high temperature.

Table V (middle) Relation of grease oil viscosity to limiting low temperature torque.

Table VI (bottom) Norma-Hoffman test significance[13].

USED GREASE EVALUATION: When used judiciously, the many tests discussed here will provide a good idea of the utility of a grease. Actual use, however, often presents conditions which are unusual or unforeseen so that even a wise choice does not guarantee long life. Furthermore, even more standard conditions do not permit us to guess exactly the life of a grease in operation. Therefore, it is desirable to have some test for the life remaining in a grease in operation. Such a test must, of course, be made with a small quantity of grease such as may be extracted from a bearing in use. Two possibilities are oil content and acidity measurements.

OIL CONTENT: The determination of the oil content of a used grease is a relatively simple operation requiring as little as one-tenth of a gram of grease[14]. In this procedure, liquids in a grease are extracted with benzene, the acids are neutralized, and the resulting soaps are removed by washing an ether solution of the oil with alcohol. The solvents are then evaporated and the oil weighed.

The oil contents of a large number of greases from failed bearings from laboratory tests and field applications have been measured, and the values are found to be very largely between 50 and 60% as shown in Fig. 4. In this bar graph, the failures at 100 C., have been added to those at 150 C. It is interesting that the maximum number of failures occurs at the same oil content at both temperatures. Such analyses offer a very convenient means of determining the approach of lubrication failure.

ACIDITY: A second measure of deterioration in a grease is a determination of the acid derived from the oxidation of the oil. The standard analysis for acidity of oils[15] must be modified to avoid interference in the titration by the soap. Therefore, the oil with its acids is first extracted from the grease with ether. The ether solution is then diluted with alcohol and a little water and titrated with alcoholic base. The acid numbers, that is, milligrams of potassium hydroxide per gram of grease, found in samples from failed bearings have been found to run in the region of 5 to 15. Not enough data are yet available to set a limiting value for bearing life. It may be that no such value will be found because a ball bearing will run even at high acid numbers, so long as free oil is present.

Future possibilities for used grease analysis exist in spectra study. The presence of metallic impurities can be detected readily by arc spectroscopy, and perhaps ultra-violet or infra-red studies can be used for identification of deterioration products.

SUMMARY: In providing satisfactory lubrication, grease must accomplish to a greater or less degree many functions. Such functions usually require at least a number of the following properties: (1) Optimum bleeding and feeding characteristics, (2) Oxidation stability, (3) Low evaporation, (4) Maximum stability against working, (5) Corrosion prevention, (6) Low torque and good pumpability characteristics, (7) Ability to exclude foreign materials, (8) Ability to channel, (9) Good adhesive characteristics, (10) Resistance to water washing.

No single bench test will characterize a lubricant as well as an actual performance evaluation, but objective use of results obtained from the tests discussed in this paper will greatly aid in selection of a suitable grease for the job.

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Section News

(SECTION SECRETARIES, please note: All Section News copy, photographs, summaries, and Personals should be submitted to ASLE National Office 1½ months prior to month of LE publication.)

CHICAGO: The complete program for the 1953-1954 Chicago Section monthly meetings has been announced as follows: Oct. 15, Stump The Experts; Nov. 19, "Economics of Petroleum;" Jan. 21, "Refrigeration Oils;" Feb. 18, Ladies Night; Mar. 18, "Lubrication Of Automotive Diesels & Aircraft Engines;" Apr. 15, "Synthetic Lubricants;" May 20, "Machining Stainless Steel & Titanium;" June 17, Golf Party. (Submitted by W. P. Green, Sec'y-Treas.)

CINCINNATI: August meeting—Annual election of officers (See ASLE Directory. Submitted by R. L. Peters, Sec'y-Treas.)

DETROIT: July meeting—Annual election of officers (See ASLE Directory).

September meeting — R. L. May, Sinclair Research Lab., presented a paper entitled "The Refining Of Lubricating Oils."

Oct. 26, meeting — Panel discussion entitled "Problems Encountered With Drawing Compounds," with W. J. Wojtowicz, H. A. Montgomery Co., Inc., F. M. Tousley, Briggs Mfg. Co., and W. E. Dewey, Ford Motor Co., as Panelists, and E. S. Kelly, The Budd Co., as Moderator. (Submitted by M. L. Beardslee, Sec'y-Treas., and T. Cabba, Program Chrmn.)

KINGSPORT: June meeting — Chicken dinner, prepared and served by members of the Mead Corp., followed by two films in connection with the Cincinnati Milling Machine Co., entitled "Physics Of Metal Cutting," and "Highways To Production." (Submitted by J. E. Fleenor, Sec'y-Treas.)

NORTHERN CALIFORNIA: August meeting — Annual election of officers (See ASLE Directory).

September meeting — D. H. Moreton, Douglas Aircraft Co., Inc., presented a paper entitled "A Review Of Synthetic Lubricants," covering the following types of synthetic lubricants: monobasic acid esters, dibasic acid esters, phosphate esters, silicone polymers, silicate esters, polyglycol ethers, and fluorinated and chlorinated compounds. Also included were such points as: the economic picture of synthetic versus mineral oil lubricants, the comparison of lubricating qualities, the effects on packings, gaskets and paint, the interest of the military in synthetic oils, and toxicological aspects. (Submitted by R. H. Decker, Sec'y-Treas.)

PHILADELPHIA: The Franklin Institute Laboratories for Research & Development, in cooperation with the Philadelphia Section of ASLE, are sponsoring a special evening lecture course (Mondays, Sept. 28 thru Dec. 14), "Advanced Lubrication," at the Franklin Institute, Benjamin Franklin Pkwy. at 20th St., offered as a service to industry to bring the most recent developments in lubrication engineering to the scientists and engineers who must apply them every day. The series will be presented by Prof. D. D. Fuller, of Columbia Univ., who will discuss from both the theoretical and practical standpoints the problems that face engineers who design and maintain the vital and complex machines used by modern industry. A postgraduate course, accredited as such by the Univ. of Delaware, the basic fee for the course is $25; an additional charge is made if the registrant wishes to have the course credits entered in the records at the Univ. of Delaware: $5 if the registrant is already enrolled at the University, $10 if he is not. (Submitted by M. E. Dougherty, ASLE Eastern Vice-President.)

TWIN CITIES: July meeting — J. W. Vollentine, Caterpillar Tractor Co., presented a paper on "The Development Of Heavy Duty Lubricating Oil Specifications" and "The Engine Tests Concerned With These Specifications," in which details of the L-1 and L-3 Engine Tests originally developed by the Caterpillar Tractor Co. were explained. (Submitted by C. D. Johnson, Sec'y.)

Coming Events

OCTOBER

19-20 Ind. Petroleum Assn. of America (annual meeting), Texas Hotel, Ft. Worth, Tex.

19-23 National Safety Council (annual congress), Conrad Hilton, Congress, Morrison, & Sheraton Hotels, Chicago, Ill.

26-28 National Lubricating Grease Inst. (annual meeting), Edgewater Beach Hotel, Chicago, Ill.

28-29 Ind. Oil Compounders Assn. (annual meeting), Edgewater Beach Hotel, Chicago, Ill.

29-30 National Industrial Conference Board, Waldorf-Astoria Hotel, New York, N. Y.

NOVEMBER

2-4 Socy. of Automotive Engineers (national transportation meeting), Conrad Hilton Hotel, Chicago, Ill.

2-4 American Oil Chemists' Socy. (27th Fall meeting), Sherman Hotel, Chicago, Ill.

3-4 Socy. of Automotive Engineers (national diesel engine meeting), Conrad Hilton Hotel, Chicago, Ill.

5-6 Socy. of Automotive Engineers (national fuels & lubricants meeting), Conrad Hilton Hotel, Chicago, Ill.

9-12 American Petroleum Inst. (33rd annual meeting), Conrad Hilton & Palmer House Hotels, Chicago, Ill.

19-21 American Standards Assn. (35th annual meeting), Waldorf-Astoria Hotel, New York, N. Y.

29 to Dec. 4 American Socy. of Mechanical Engineers (annual meeting), Statler Hotel, New York, N. Y.

30 to Dec. 5 24th Exposition of Chemical Industries, Grand Central Palace, New York, N. Y.

APRIL 1954

5-7 American Society of Lubrication Engineers (9th Annual Meeting & Exhibit), Netherlands Plaza Hotel, Cincinnati, Ohio.

THE FATIGUE SPIN RIG—

A New Apparatus For Rapidly Evaluating Materials & Lubricants For Rolling Contact*

by E. F. Macks
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INTRODUCTION: Rolling contact under elastic deformation causes fatigue failures which limit the useful life of ball and roller bearings, gears, and other machine elements.

Fatigue of rolling-contact bearing elements has received extensive attention[1 to 16]. Gear manufacturers and others have also studied the fatigue problem vigorously[17 to 41]. There is, however, relatively little published literature on the endurance life and fatigue strength of alloy steels hardened to Rockwell C-60 or greater[42 to 47]. While no attempt has been made to include a complete bibliography on fatigue, certain significant references have been cited; and in particular, some recent studies which relate to the subject problem are included.

Rolling-contact-bearing life ratings are being continually increased because of better manufacturing techniques, and closer material and inspection control. However, the life dispersion of supposedly identical bearings has not been improved significantly. This dispersion still varies 5000 percent or more[47].

The fatigue life of a machine element undergoing fluctuating stress is generally considered to be a function of the number and magnitude of the stress fluctuations. However, bearing fatigue data show a ratio of maximum to minimum life of 40 or more when plotted in this manner, while simple specimens in bending show this ratio to be in the neighborhood of seven[31]. Furthermore, the mean life of the simple specimens is about 2.5 times the minimum life; while for bearings, the mean life is 10 or more times the minimum life. The wide dispersion of fatigue data, and particularly of bearing fatigue data, when regarded only on the basis of number and magnitude of stress fluctuations, suggests: (a) that factors other than number and magnitude of stress fluctuations influences bearing fatigue life, or/and (b) that it has not been possible to reproduce the magnitude of the significant stresses with the precision required to restrict the scatter. The magnitudes of the significant stresses may vary, in supposedly identical tests, due to external factors such as non-reproducible test conditions and tolerances in bearing geometry, etc., or due to internal factors such as anisotropic properties within the stressed volume. Such non-homogeniety might be influenced by grain direction due to processing, residual stresses due to heat treatment and finishing operations, segregation, and minute inclusions.

Long periods of time are required for bearing fatigue investigations by conventional methods. Attempts to conserve time and expense in bearing fatigue testing have resulted in a number of bench tests[42 to 44 & 47 to 54]. In achieving simplicity, such tests generally impose stresses and other conditions unlike those occurring in practice. Bearing companies consequently test 50 to 100 "identical" full-size bearings to determine the effect of each variable. The scatter of results, due in part to the imponderable variables inherent with full scale bearing tests taxes statistical methods to reveal the effects of individual variables. Although it is fully realized that complete bearing tests followed by full-scale tests in the actual application constitute the final phases in any investigation, the saving in time, effort and cost which a comparison test may offer makes any such new technique worthy of consideration.

The urgency of the problem is illustrated by the fact that the life of rolling-contact thrust bearings is a critical limitation of aircraft-gas-turbine design. The loads at the bearing-surface speeds involved are far beyond existing bearing knowledge. For maintenance of reasonable bearing life and reliability at the high temperatures to which the bearings are subjected, materials are required which possess good high-temperature fatigue-life properties as well as corrosion resistance. Little is known about either the room-temperature or high-temperature fatigue life of the majority of materials from which many experimental bearings for high temperature operation are now being made.

It is the object of this paper to describe a simple apparatus which, it is believed, will permit rapid and valid evaluation of the effect of independent variables on the fatigue life of balls and raceways over a wide temperature range. A further objective is to illustrate the possible utility of the apparatus for lubricant studies and for fundamental research in rolling contact.

APPARATUS. GENERAL DESCRIPTION: The fatigue spin rig is shown in Fig 1. In the main, the apparatus consists of a raceway A which is resiliently mounted in the horizontal plane on toothpick springs B, a vertically adjustable nozzle assembly C, and a vertically adjustable bottom plate D. Pressurized air, entering at E, causes two or more balls to rotate within the raceway. The spent air is allowed to escape at F. The speed of ball rotation is controlled by adjusting the pressure of the air entering at E.

THEORY OF OPERATION: The subject apparatus is functional above the critical frequency of the system because of the following factors: (a) The rotating balls space themselves to provide dynamic balance. (b) For a given test, ball rotation is maintained over precisely defined ball and raceway contact areas.

An analysis of the stability characteristics of the fatigue spin rig is given in the appendix where it is shown for the general case of two balls of unequal mass that (A8):

$$r_p = \frac{\left[\frac{m_1}{m_2}\cos\alpha + \sqrt{1-\left(\frac{m_1}{m_2}\sin\alpha\right)^2}\right]\left(\frac{\omega}{\omega_n}\right)^2}{\frac{M}{m_2(r_t-r_b)}\left[1-\left(\frac{\omega}{\omega_n}\right)^2\right]}$$

where r_p is the radius of raceway precession and the other symbols are defined in the appendix. Smooth operation occurs if r_p is very small (for the practical case a few thousandths of an inch). This condition is satisfied above the critical frequency of the system if m_1/m_2 approaches unity. In such a case, it is shown that a pair of balls rotate essentially at 180° and equilibrium operation will prevail.

It is further shown in the appendix that for the case of a pair of balls of essentially equal mass rotating within a balanced raceway assembly, that is, when the center of gravity is coincident with the geometric center of the raceway assembly, the coefficient of rolling friction μ may be determined experimentally by measurement of the angle θ and by use of the expression (A12):

$$\mu = \frac{MR^2\theta}{2r_t(r_t-r_b)(m_1)\left(\frac{\omega}{\omega_n}\right)^2}$$

where θ is so calibrated as to eliminate factors other than rolling friction and where the symbols are again defined in the appendix.

It is also shown that three balls of equal mass will not operate in equilibrium; however, for the case where two balls are of equal mass and the third ball is of different mass equilibrium operation may be expected.

For the case of four balls of equal mass the discussion following equation (A8) applies, and the balls may be expected to rotate in equilibrium in pairs at essentially 180°. For the case of five balls of equal mass the balls may be expected to rotate in equilibrium in two pairs and a free ball.

It has been observed visually by means of a stroboscopic light source and transparent rig components that the analysis accurately predicts the operating stability characteristics of the fatigue spin rig.

*Sponsored by the ASLE Technical Committee on Bearing Lubrication and presented at the ASLE 8th Annual Meeting, Boston, April 14, 1953.

Under proper conditions of operation, the balls lift from the bottom plate to ride continuously at an equilibrium position on a vertical raceway wall. Acceleration, gyroscopic, and aerodynamic forces apparently provide the lifting and required stabilization. An analysis of this phenomenon was unsuccessful because of the unknown aerodynamic forces.

Inasmuch as a pair of balls, or a trio of balls of proper mass ratio, rotate in dynamic equilibrium; and since the balls do not touch any surface except the raceway wall, the loads at the respective contact areas between the balls and the cylinder may be determined with precision. These loads are caused by the centrifugal forces resulting from the orbital motion of the balls about the raceway axis.

INSTRUMENTATION: Automatic operation, together with the accumulation of precise data, may be accomplished by employing a vibration pickup, an electronic tachometer and scaler, and an automatic speed regulator. An automatic shutoff circuit may be incorporated to shut off the rig and the desired instrumentation when a small pit develops.

RESULTS: Preliminary experiments showed that the fatigue failures produced with the fatigue spin rig were of the type that occur in rolling-contact bearings in practice.

RELATIVE MOTION: Rolling motion under elastic deformation occurs between the surfaces in contact. A representative ball path obtained in the fatigue spin rig is shown in Fig. 2. The ball path shown was obtained by rotating a pair of one-half inch diameter chrome steel balls at an orbital speed of 28,000 rpm within a 3.310 inch bore, ungrooved tool steel raceway for about 7 minutes which resulted in 1,358,000 ball cycles. No lubrication was provided. The primary rolling lines and sliding contact bands associated with rolling under elastic deformation are clearly visible. Similar contact bands occur on the raceway. The width of the complete contact band as calculated (0.038 inch approx.) checks favorably with experimental findings (0.041 inch approx.), the experimental values being about eight percent higher.

STRESSES: The stresses produced in the subject apparatus bear the same relation to load as do the stresses in a loaded ball bearing. Surface compressive and tangential stresses and subsurface shear stresses are imposed at the contact area and within the stressed volume, respectively.

Stress frequencies to over 10 million cycles per hour and stress magnitudes to over 700,000 pounds per square inch (calculated, Hertz) have been obtained. For the path shown in Fig. 2, the maximum calculated surface pressure was 735,000 pounds per square inch. The magnitudes of the imposed stresses are varied by adjustment of the drive-air pressure to change the speed of ball rotation. The stresses can be varied at a given frequency of application by changing specimen geometry.

To illustrate the ranges over which certain significant variables may be investigated by means of the fatigue spin rig for a specific case, the following calculations have been made: centrifugal force (P_0); the major and minor contact ellipse axes (2a) and (2b) respectively, (2b is equal to the contact band width which may be measured on the ball or raceway); maximum surface compressive pressure (S_{max}); distance of maximum shear stress below the surface (z); and approach of the ball and raceway (δ). The methods of Hertz[55], Thomas and Hoersch[56], Belajeff[57], Odquist and Lundberg[58], and Jones[59] have been employed in the stress and deformation calculations. The values of the foregoing significant variables are plotted in Fig. 3 for a 3.25-inch-bore ungrooved steel raceway over an orbital speed range (N) from 10,000 to 30,000 rpm with ball diameter ($2r_b$) as parameter. The values of mean compressive stress and the maximum subsurface shear stress may be estimated from figure (3d) where the maximum shear stress is equal to approximately 0.325 S_{max} and the mean surface compressive pressure is equal to 2/3 S_{max}. Various assumptions have necessarily been made in the calculations on which the results of Fig. 3 are based. Whereas the above-mentioned investigators[55 to 59] did not include tangential surface loading, Mindlin[60] includes consideration of a small tangential force and a small torsional couple across an elliptic contact surface for the case where the nonslip area is symmetrical with respect to the contact area. Cattaneo[61] has also investigated the problem of the effect of tangential force on an elliptic contact surface, and Poritsky[62] studied this effect for a special case of cylindrical bodies in contact. The nonsymmetric orientation of these areas, as for ball bearings under combined load, apparently remains to be investigated. Therefore, Fig. 3 gives only as a first approximation the magnitudes of the significant variables associated with operation of the subject apparatus. Similar charts may be prepared for practically any geometric configuration.

Inasmuch as the fatigue life of a complete bearing varies inversely with the ninth to tenth power of the contact stress[9], the importance of stress reproduction in fatigue tests is evident. By using the following relation, precise contact loads may be obtained with the subject apparatus if the mass (m) and the centrifugal acceleration of the balls rotating about the raceway axis are known (1): Load = m a.

As an example of the precision to which contact loads may be determined and reproduced from test to test in the fatigue spin rig, the following values are cited: Steel balls ½ inch in diameter and of grade-1 precision weigh within 0.030 gram of each other. This variation in sets of balls gives rise to about 0.36 percent variation in resultant ball load. Greater accuracy may be obtained by matching ball weights to within limits as close as 0.0001 gram. Variations in orbital speed of the balls about the raceway axis may be held to within $\pm$0.2 percent by controlling drive-air pressure. This results in a maximum variation in ball load of $\pm$0.6 percent for a ½-inch-diameter ball set rotating within a 3.25-inch-bore raceway at 25,000 rpm. A first approximation in maximum variation in life based upon the variation of orbital ball speed and matched ball weight (within 0.0001 gram) is then about 6 percent. Such precision in control of external test variables is considerably better than can be obtained when testing complete bearings.

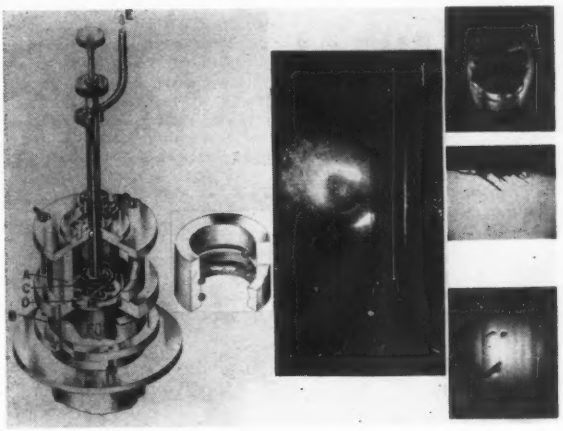


Fig. 1 (left) Fatigue spin rig.

Fig. 2 (center) Representative ball path obtained in fatigue spin rig. Ball cycles, 1.358×10^3; maximum compressive surface pressure (Hertz), 735,000 psi.

Fig. 4 (right) Typical fatigue failures obtained in fatigue spin righ. (top) Top view; (middle) Section view (a) Ball cycles, 34.2×10^3; maximum compressive surface pressure (Hertz), 654,000 psi; duration of test, 224 minutes; (bottom) (b) Ball cycles, 30.6×10^6; maximum compressive surface pressure (Hertz), 648,000 psi; duration of test, 199 minutes.

TYPICAL FATIGUE FAILURES: Top and sectional views of a ball fatigue failure obtained in the fatigue spin rig are shown in Fig. 4a. The fatigue pit is seen to be slightly larger than, and centered with respect to, the ball path. The failure occurred after 34.2×10^3 ball cycles at 654,000 pounds per square inch maximum compressive surface pressure (Hertz). The test consisted of rotating a pair of ½-inch-diameter chrome-steel balls at an average orbital speed 23,350 rpm within an ungrooved 3.250-inch-bore tool-steel cylinder for 224 minutes.

A fatigue failure which was caught in the early stage is shown in Fig. 4b. The test was of 199 minute duration. During this interval 30.6×10^3 ball cycles at 648,000 pounds per square inch maximum compressive surface pressure were imposed by rotating a pair of ½-inch-diameter chrome-steel balls at an average orbital speed of 23,100 rpm within an ungrooved 3.330-inch-bore tool-steel cylinder.

APPLICATIONS OF APPARATUS: The fatigue spin rig has limitations as well as several unique features and a wide range of application. Some of the many uses of the rig are mentioned in the following paragraphs to illustrate its versatility.

GROOVED OR PLAIN RACEWAY PATHS: The raceway path may be plain or grooved (Fig. 1).

Grooves of the type used in deep-groove ball bearings may be employed. The groove radius may be varied to study the effect of the ratio of major to minor contact ellipse axes, as well as other factors.

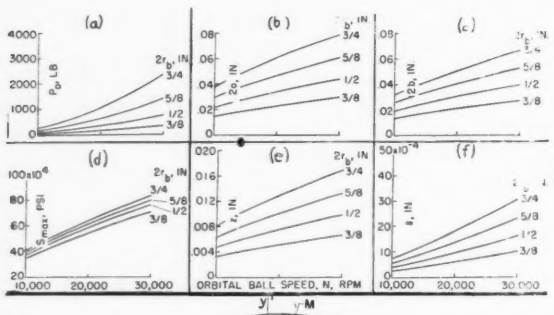
A force component at the contact area similar to that occurring in an angular contact bearing under combined radial and thrust load may be introduced by using V-type raceway grooves.

RACEWAY WITH SEVERAL PATHS: A single raceway may be used for several tests. For example, a raceway may be equipped with several grooves along its axis. A raceway without grooves may be used for many tests merely by causing the balls to rotate at various vertical locations. This may be accomplished by means of the adjustable nozzle and the bottom-plate assembly (Fig. 1). This feature not only simplifies testing and reduces the cost thereof but it also eliminates a significant variable.

INTERRUPTED TESTS: If a small hole is drilled through the diameter of each ball in a set, gyroscopic forces will cause the balls to rotate about the centerline of the hole[63]. In order to eliminate windage effects, the hole may be filled with a material having a density different from that of the ball material. Therefore a test may be stopped at any time, the balls examined or processed, and the test started again with assurance that the same path will be retraced on each ball.

This feature also makes it possible to reduce test time appreciably. For example, in testing a pair of identified balls a failed ball may be replaced by a new ball. The remaining ball is thereby allowed to build up stress cycles over a precisely defined contact area as the procedure is continued until the desired number of ball failures is reached. Use of a new raceway path for each test would depend upon the test conditions and the judgment of the investigator. If it is desired to study the raceway as well as the balls, the same raceway path would be used until a failure on it occurred. In this manner ball and raceway data can be rapidly accumulated over precisely defined areas and under loads which are precisely known.

An alternate method of causing raceway failure is to replace sets of balls before ball failure occurs and thereby to continue building up stress cycles on the raceway path until a raceway failure occurs.



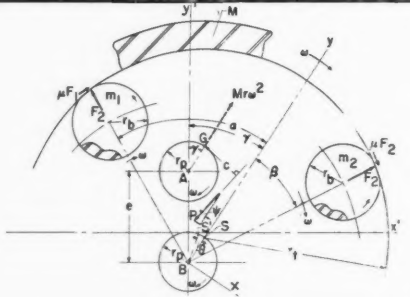


Fig. 3 (top) Significant variables as function of orbital ball speed with ball diameter as parameter. Ungrooved, 3.25-inch-bore steel raceway and steel balls.

Fig. 5 (bottom) Fatigue spin rig equilibrium force diagram on raceway in plane of ball rotation. Two balls of unequal mass but essentially of equal size.

COEFFICIENT OF ROLLING FRICTION: The coefficient of rolling friction may be determined over a wide range of operating conditions and for many operating variables.

DYNAMIC LOAD EFFECT: It has been found that three balls of essentially equal weight will not rotate in dynamic equilibrium (see appendix). However, a set of two light balls and one heavy ball or a set of two heavy balls and one light ball will rotate in equilibrium. If a grooved raceway path is employed, the effect of superimposed loads of various magnitudes and contact intervals may be studied.

LUBRICANT STUDIES: A grooved raceway which is split may be used for lubricant research. For example, each half of a "V" grooved path, which is split and insulated at the apex of the V, will be electrically connected by the rotating balls. The electrical properties of such a circuit may be expected to vary widely because of the presence or lack of a lubricant film between the metal surfaces. The effect of lubricant viscosities, additives, etc., upon film strength may then be determined over a wide range of operating conditions. In this manner it may also be possible to study the effect of rate and magnitude of loading on the transition from hydrodynamic to boundary lubrication. Such investigations are significant regarding gear, as well as lubricant and bearing research. Care must be exercised in the interpretation of results obtained in this manner as different lubricants may have different electrical properties even for the same film thickness. Also, the electrical properties of certain lubricants may change with shear rate, as well as with operating temperature.

Inasmuch as the contact areas between the balls and the raceway are the only surfaces undergoing relative motion, investigations may be made either completely dry or lubricated.

HIGH- OR LOW-TEMPERATURE OPERATION: Studies of fatigue, wear, and lubrication, as well as fundamental studies of rolling and sliding friction, may be made over a temperature range limited only by the temperature to which the required quantity of drive-air may be heated or cooled. Auxiliary equipment may be employed to heat or cool the raceway.

VISUAL & PHOTOELASTIC STUDIES: Visual and photoelastic-stress investigations may be made by employing transparent rig components and a polarized, stroboscopic light source in phase with the events to be studied.

COMPARISON STUDIES: One of the main uses of the rig is for comparitive-type tests to determine the relative effects of independent variables. A few such comparisons are: materials, operating temperatures, heat treatments, manufacturing methods, surface finishes, platings, coatings, and lubricants.

SUMMARY: An apparatus is described which permits independent evaluation of many variables that affect the fatigue life of bearing balls and raceways. Precise loads at very high frequencies over a wide range of variables are possible. Stress frequencies to 10 million cycles per hour and stresses to 700,000 pounds per square inch (calculated, Hertz) have been obtained. Rolling contact of the specimens provides surface compressive tangential, and subsurface shear stresses which induce fatigue failures of the type occurring in rolling contact bearings under radial or combined loads.

The apparatus is adaptable to very high- or low-temperature operation. Investigations of the effect of lubricant properties on endurance of materials and studies of lubricants for film strength and other properties may be made. Fundamental investigations may be made over wide ranges of variables.

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APPENDIX—SYMBOLS (Refer to Fig. 5)

A	location of center of gravity of raceway assembly when at rest
B	center of raceway when at rest.
c	moment arm of P about G
e	distance between S and G (eccentricity), in.
F_1, F_2	centrifugal inertial force acting on raceway due to orbital rotation about B of balls of mass m_1 and m_2, respectively, lb
G	center of gravity of raceway assembly
k	horizontal spring constant of resilient raceway mount (three toothpick springs at 120°) in any direction in plane of ball rotation, lb/in.
M	mass of raceway assembly (not including balls), lb-sec²/in.
m_1, m_2	mass of lighter and heavier ball respectively, lb-sec²/in.
P	restoring force, lb
R	radial distance from S to center of each of three toothpick springs, in.
r_b	radius of each ball, in.
r_p	radius of precession of S about B and G about A, in.
r_t	radius of raceway track, in.
S	center of raceway in plane of ball rotation when raceway precesses (B and S coincide when raceway is at rest)
x, y	coordinate axes chosen in accordance with Fig. 5 where the y-axis is always colinear with points B and S
x', y'	particular coordinate axes denoting when B, S, and G are co-linear and with S above B
α	angle between y-axis and centerline of lighter ball, deg
β	angle between y-axis and centerline of heavier ball, deg

γ	angle between y and y'
θ	rotational displacement of S' to S due to friction forces, deg
μ	coefficient of rolling friction
ΣF_x	summation of forces in x-direction
ΣF_y	summation of forces in y-direction
ΣM_G	summation of moments about center of gravity of raceway assembly
ψ	angle between P and y-axis, deg
ω	angular velocity of balls, G, S, and coordinate x, y-axes about B, rad/sec
ω_n	critical angular velocity, rad/sec

ANALYSIS

GENERAL CASE—TWO BALLS: Consider two balls of unequal mass to rotate in a condition of equilibrium in the horizontal plane within a raceway which is resiliently mounted in the plane of ball rotation, Fig. 5. For two balls of unequal mass but essentially of equal size, the following equilibrium force and moment relations exist (A1): When $\Sigma F_x = 0$, then $F_2 \sin\beta + \mu F_2 \cos\beta + \mu F_1 \cos\alpha = F_1 \sin\alpha + P \sin\psi$, (A2): When $\Sigma F_y = 0$, then $F_1 \cos\alpha + \mu F_1 \sin\alpha + Mr_p\omega^2 + F_2 \cos\beta = \mu F_2 \sin\beta + P \cos\psi$, (A3): When $\Sigma M_G = 0$, then $Pc - F_2 [e \sin(\beta+\gamma) + r_p \sin\beta] + F_1 [e \sin(\alpha-\gamma) + r_p \sin\alpha] + \mu F_1 [r_t - e \cos(\alpha-\gamma)] + \mu F_2 [r_t - e \cos(\beta+\gamma)] = 0$.

Substitution for F_1 and F_2 in equation (A2) yields the following since $P \cos\psi = kr_p$, (A4): $m_1 (r_p \cos a - r_b + \sqrt{r_t^2 - (r_p \sin\alpha)^2})(\cos\alpha + \mu \sin a)\omega^2 + Mr\omega^2 + m_2 (r_p \cos\beta - r_b + \sqrt{rt^2 - (r_p \sin\beta)^2})(\cos\beta - \mu \sin\beta)\omega^2 = kr_p$.

It is known that for smooth operation r_p must be very small, i.e., in the neighborhood of a few thousandths of an inch. Therefore, since $(r_p \sin\alpha)^2 << r_t^2$, $r_p \cos\alpha << (r_t - r_b)$, $(r_p \sin\beta)^2 << r_t^2$, and $r_p \cos\beta << (r_t - r_b)$, equation (A4) reduces to (A5): $m_1 (r_t - r_b)(\cos\alpha + \mu \sin\alpha)\omega^2 + Mr_p\omega^2 + m_2 (r_t - r_b)(\cos\beta - \mu \sin\beta)\omega^2 = kr_p$. Also, since $\mu F_1 << F_1$ and $\mu F_2 << F_2$, (A6a):

$$\sin\beta = \frac{m_1}{m_2} \sin\alpha, \text{ (A6b):}$$

$$\cos\beta = \sqrt{1 - \left(\frac{m_1}{m_2} \sin\alpha\right)^2}.$$

If equations (A6) are substituted into equation (A5) and the result is solved for r_p, (A7):

$$r_p = \frac{\left[m_1(r_t - r_b)(\cos\alpha + \mu \sin\alpha) + m_2(r_t - r_b)\left(\sqrt{1 - \left(\frac{m_1}{m_2} \sin\alpha\right)^2} - \mu \frac{m_1}{m_2} \sin\alpha\right)\right]\omega^2}{k - M\omega^2}$$

and since $\omega_n^2 = \frac{k}{M}$, (A8):

$$r_p = \frac{\left(\frac{m_1}{m_2} \cos\alpha + \sqrt{1 - \left(\frac{m_1}{m_2} \sin\alpha\right)^2}\right)\left(\frac{\omega}{\omega_n}\right)^2}{\frac{M}{m_2(r_t - r_b)}\left[1 - \left(\frac{\omega}{\omega_n}\right)^2\right]}$$

The stability characteristics of the fatigue spin rig may be established by means of equation (A8). Since smooth operation results when r_p approaches zero, the conditions under which r_p can approach zero must be found. It is apparent that: (1) For $\frac{\omega}{\omega_n} < 1$, $r_p >> 0$ since the balls tend to rotate about B at the largest possible radius; they therefore locate on the heavy side of the system. (2) For $\frac{\omega}{\omega_n} = 1$, $r_p \to \infty$, a critical frequency. (3) For $\frac{\omega}{\omega_n} > 1$, the tendency is for the balls to locate on the light side of the system; they therefore try constantly to rotate about B at the smallest possible radius. When $m_1 \neq m_2$ the value of r_p cannot be zero because of an unbalanced exciting force. For the special case when $m_1 = m_2$, r_p reaches zero at $\alpha = \pi/2$ (see equation (A8)). If the balls go past the balance point ($\alpha > \pi/2$), they immediately find themselves on the heavy side since r_p is positive. They instantaneously tend to locate on the light side making $\alpha < \pi/2$ and r_p negative which again places them on the heavy side. Therefore, complete stability results only when $r_p = \cos\alpha = 0$.

BALANCED RACEWAY ASSEMBLY — TWO BALLS: For the particular case of two balls of essentially equal mass rotating essentially at 180° in a balanced raceway assembly, that is, when G and S are coincident (e is zero), the restoring force P will be in a direction opposite that shown in Fig. 5, and equation (A3) reduces to the following, (A9): $\mu F_1 r_t + \mu F_2 r_t = Pc$. Therefore, for the type of resilient mount shown in Fig. 1,

$$2m_1\mu r_t \omega^2\left(r_p \cos\alpha - r_t + \sqrt{r_t^2 - (r_p \sin\alpha)^2}\right) = kR^2\theta$$

Simplification yields (A10):

$$\mu = \frac{kR^2\theta}{2r_t(r_t - r_b)(m_1)\omega^2}$$

or (A11):

$$\mu = \frac{MR^2\theta}{2r_t(r_t - r_b)(m_1)\left(\frac{\omega}{\omega_n}\right)^2}$$

It is apparent from equation (A11) that by measurement of the calibrated angle θ an experimental determination of the coefficient of rolling friction is possible over a wide range of operating conditions.

THREE BALLS OF EQUAL MASS: For the special case of 3 balls of equal mass (m), and with the y-axis passing through one ball and each adjacent ball thereto spaced at α, (A12):

$$r = \frac{(1+2\cos\alpha)\left(\frac{\omega}{\omega_n}\right)^2}{\frac{M}{m(r_t - r_b)}\left[1 - \left(\frac{\omega}{\omega_n}\right)^2\right]}$$

(1) For $\frac{\omega}{\omega_n} < 1$, $r_p >> 0$. (2) For $\frac{\omega}{\omega_n} = 1$, $r_p \to \infty$, a critical frequency. (3) For $\frac{\omega}{\omega_n} > 1$, that is, above the critical frequency, r_p tends to be negative, which means that $\cos\alpha > (-0.5)$; however, this automatically places the balls such that their resultant centrifugal force acts on the side on which r_p is greatest. (Assume $\alpha = 120°$ at the start of the foregoing discussion.) The balls immediately try to move to the side where r_p is smallest (the light side of the system). When each ball is of the same mass and therefore of the same inertia, the balls move instantaneously; and two balls always move in the same direction, while the third ball moves in the opposite direction. Thus, three balls of equal mass cannot be expected to reach equilibrium.

THREE BALLS — TWO OF EQUAL MASS: For the case of three balls where two are of equal mass and the third is of different mass, equilibrium operation may be expected. With the balls so located that their resultant centrifugal force acts on the heavy side, they immediately try to move to the side on which r_p is smallest. However, because of lower inertia, the lighter ball (or balls) will move to the light side before the heavier ball (or balls) does. As the light ball (or balls) passes through the balance point, r_p changes sign instantaneously and the light balls again find themselves on the heavy side and the process is repeated before the heavier balls can overcome their higher inertia. The light ball (or balls) thus provides the stability for a three ball system where two balls are of equal mass.

FOUR OR MORE BALLS: For the case of four balls of equal mass, the analysis following equation (A8) applies, and the balls may be expected to rotate in equilibrium in pairs essentially at 180°.

For the case of five balls of equal mass, the analysis following equation (A12) applies, and the balls may be expected to rotate in equilibrium in two pairs and a free ball.

In general, the analysis following equation (A8) may be expected to hold for even numbers of balls of equal mass, while the analysis following equation (A12) may be expected to hold for odd numbers of balls of unequal mass.

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SCORING & WEAR OF SPUR GEARS*

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INTRODUCTION: During the last decade the literature on medium and high speed gears and their use in the transmission of power has included many papers on gear lubrication. This shows that the importance of proper gear lubrication and even the desirability of including the lubricant as a design factor are now well recognized, since lubrication failures can be just as destructive as mechanical failures.

Although no general agreement on the mechanism of gear lubrication prevails as yet, practically all investigators agree that scoring, also called scuffing, is the first and the most destructive type of gear lubrication failure. They also agree that scoring occurs when the lubricating film between the contacting surfaces breaks down.

Regarding the nature of this film two schools of thought exist at present. One, represented by Blok[1], Lane[2], Hutt[3] and Kelley[4], teaches that the oil films between gear teeth is thin, i.e., lubrication is of the boundary type in which the viscosity of the lubricant and hydrodynamic forces play only a negligible role. Under such conditions a high temperature flash occurs at the moment and point of contact. The boundary film will fail and scoring begin, if the flash exceeds a critical temperature which is assumed to be a characteristic property of each lubricant.

The other school, represented by Gatcomb[5], Cameron[6], McEwen[7], and others, believes that in the operating range of most gears hydrodynamic forces play the major role, i.e., the mating gears are separated by a thick lubricating film. This is considered the case wherever straight mineral oils are used that contain no chemically reactive components. Under these conditions the viscosity of the lubricant would be its most important property from the viewpoint of load carrying capacity. Equations for the calculation of these hydrodynamic forces were initially derived from the analysis of the contact of two rotating disks with the assumption of rigidity of the disks, constant oil viscosity, negligible inertia forces, constant load and steady state conditions. The values obtained from these equations were far below those measured in load-speed-friction tests on disk machines, and the investigators mentioned soon demonstrated that the effect of deviations from the assumptions made could well account for the large differences. When the assumptions are dropped, however, the resulting actions and interactions are very complex and have remained largely unassessed as yet.

These theories, both that including and that excluding hydrodynamic effects, result in equations which, with reasonable adjustment for each particular case, describe in a general way the score limited performance of non-reactive mineral oils. However, very few data are available to prove or disprove the validity of the basic theories and almost nothing is known of their applicability to the field of reactive lubricating oil components. The complexity of the subject seriously limits what can be further accomplished by theoretical analysis alone.

In the face of this situation an empirical approach was taken at the author's laboratory, based on load carrying capacity tests carried out on actual gears operated in a conventional four square machine. This machine and the results of earlier tests on it were described in a previous presentation[8] whose significant points will be reviewed here.

THE THICK FILM & THIN FILM REGION OF LUBRICATION: The experiments were performed with highly accurate and finished spur gears of 6 diametral pitch and 20° pressure angle with an unmodified involute profile and a case hardness of 62 RC. The results with straight, unreactive mineral oils showed the existence of a relation between the power transmitted when scoring begins, and the product of lubricant viscosity and speed. This relation is shown as the horsepower-ZN curve in Fig. 1. Included on the figure are some points for lubricants compounded with E. P. agents which all fall above the curve. Microscopic inspections of all gears operated in the region below the curve showed the absence of any changes in the condition of the working surfaces. Furthermore no wear could be detected by weighing the test gears before and after such operation.

From these observations, particularly with regard to wear, it was concluded that the Hp-ZN curve divides gear operation into two regions, one below the curve where the surfaces are separated by a thick oil film and the other above the curve where boundary lubrication prevails. If the lubricant is a non-reactive mineral oil no operation is possible on or above the curve without scoring. If E. P. agents are present in the boundary lubrication region, they form a thin film which permits operation without scoring.

In presenting the Hp-ZN curve and its equation attention was called to a number of limitations. It was pointed out that the data were obtained on highly accurate and polished gears, accurately aligned and operated under steady load. Where these conditions are not met, the results could be expected to be lower than shown on the curve.

Regarding the lubricants it was stressed that the Hp-ZN relation as a definition of the scoring

*Sponsored by the ASLE Technical Committees on Lubrication Fundamentals, and Physical Properties of Lubricants, and presented at the ASLE 8th Annual Meeting, Boston, April 13, 1953.

limit is applicable only to non-reactive mineral oils. To minimize and hold substantially constant any natural E. P. activity resulting from sulfur or other impurities which even straight mineral oils may contain, the experiments were performed in a low temperature range, 30 to 50 C., with oils containing less than 0.2% sulfur. It was also demonstrated that fluids other than mineral oils do not follow the same relation due probably to that as yet unassessed "oiliness" factor.

During the two and half years since the original presentation of these empirically derived ideas on the mechanism of gear lubrication, the research work has been continued. The results which confirm the existence of the boundary and the thick film gear lubrication zones with the Hp-ZN curve as the approximate dividing line, will be presented in the following.

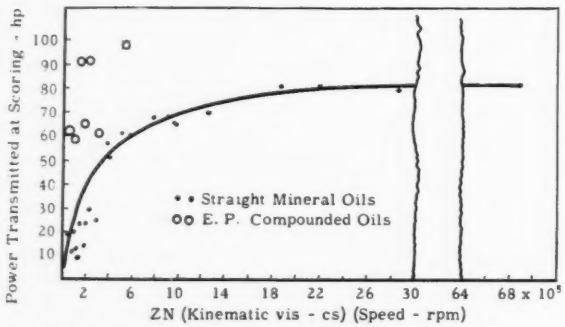


Fig. 1 Hp-ZN Curve.

EFFECT OF TEMPERATURE: As just mentioned natural or added components in particular sulfur, may become chemically active at some elevated temperature and exert an E. P. effect which can overshadow the effect of viscosity. This is demonstrated by the response of several different lubricants to temperature changes as shown in Fig. 2.

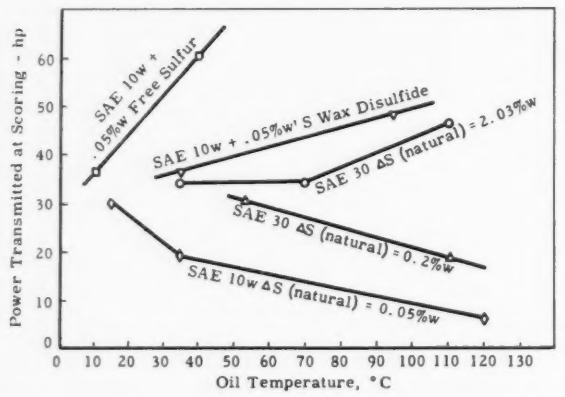
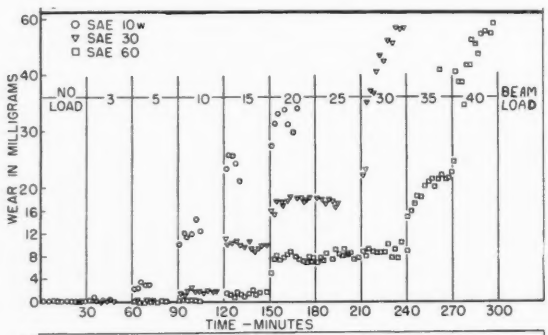

Power Transmitted at Scoring - hp
60
50
40
30
20
10
0
SAE 10w + .05%w Free Sulfur
SAE 10w + .05%w S Wax Disulfide
SAE 30 ΔS (natural) = 2.03%w
SAE 30 ΔS (natural) = 0.2%w
SAE 10w ΔS (natural) = 0.05%w
0 10 20 30 40 50 60 70 80 90 100 110 120 130
Oil Temperature, °C

Fig. 2 Effect of temperature on power transmitting capacity of oils.

The oils which are practically free of sulfur show a decrease in power transmitting capacity as temperature is increased, corresponding to the changes in viscosity with temperature. But the oils rich in natural sulfur or containing added sulfur show an increase in power transmitting capacity with an increase in temperature. Since commercial mineral oils vary widely in sulfur content as well as in the chemical nature of the sulfur compounds present, it is obvious that the effect of viscosity on scoring cannot be evaluated without taking into account the sulfur and its activity toward steel at the operating temperatures.

METAL TRANSFER & WEAR MEASURED BY RADIOACTIVE TECHNIQUES: As described earlier, the conclusion that the region below the Hp-ZN curve is that of hydrodynamic lubrication was based in part on the absence of any change in surface appearance, under microscopic examination, and on the absence of wear measurable by weighing. However, it was observed that with E. P. oils, operating without scoring above the Hp-ZN curve, the preservation of the working surfaces was often remarkably good, with only very light polishing of the asperities. Wear was also hardly measurable. Thus the difference in appearance and wear between unscored gears operated above and below the Hp-ZN curve (obviously only E. P. oil operated above Hp-ZN curve) was small, and it appeared desirable to find a new research tool to demonstrate the significant differences which would undoubtedly exist if lubrication were by thin film in one case, and by thick film in the other. This new tool was found in radioactive tracers. The technique used was fully described in a published article[1] and therefore only a brief resume will be given here.



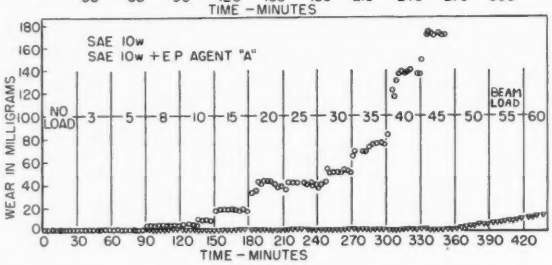


Fig. 3 Wear vs time and load; radioactive tracer technique. Pinion Speed = 3000 RPM, Beam Load x 72.12 = Load in lbs./in.

Fig. 4 Wear vs time and load; radioactive tracer technique. Pinion Speed = 3000 RPM, Beam Load x 72.12 = Load in lbs./in.

In these tests one of the two mating spur gears was radioactive. Wear was measured by placing a Geiger tube in the circulating oil stream, observing the counting rate and calculating therefrom the amount of radioactive iron in the oil. Metal transfer from the radioactive pinion to the inactive gear was evaluated by monitoring of the working surfaces of the inactive gear and by radiographing them. The tests were performed with non-reactive mineral oils of SAE 10w, SAE 30 and SAE 60 grade with sulfur content less than 0.2% and with the same SAE 10w oil compounded with a commercial E. P. agent.

The results concerning wear are presented in

Figs. 3 & 4. In Fig. 3 wear with the three straight mineral oils is plotted as a function of time at different loads. The lightest oil gave the greatest wear, and with it wear began at the lowest load. The two other oils follow in order of their viscosity. In all cases the appearance of the first sign of wear coincided with incipient scoring easily detected by visual observation.

In Fig. 4 the wear with SAE 10w straight mineral oil and with the same oil containing an E. P. agent are shown. The compounded oil shows practically no wear up to 50 pounds load. At higher loads wear increased gradually. Since no scoring occurred with this E. P. oil, it must be concluded that this wear was of a chemical nature; i.e., the boundary film which is formed by reaction between the gear metal and the additive is continually lost by attrition and reformed.

With the straight mineral oil wear did not increase gradually, but rather by jumps which occurred at the beginning of each load period above the score load. During the remainder of these periods no increase or additional wear was observed, making it probable that the initial removal of metal allowed the reestablishment of a hydrodynamic film. Only at extremely high loads wear became progressive with time. Evidently the additional removal of metal had so altered the tooth profile that the film could no longer be reestablished.

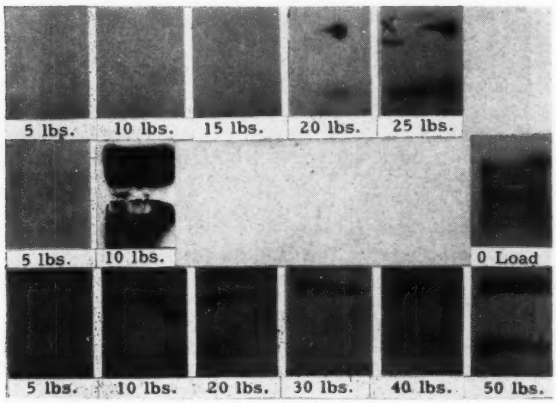


Fig. 5 Radiographs of working teeth surfaces of inert gear. (Top) SAE 60 Oil: 5 lbs., 10 lbs., 15 lbs., 20 lbs., 25 lbs. (Middle left) SAE 10w Oil: 5 lbs., 10 lbs. (Middle right) Dry: 0 Load. (Bottom) SAE 10w plus E.P. Agent: 5 lbs., 10 lbs., 20 lbs., 30 lbs., 40 lbs., 50 lbs.

A study of the metal transfer from the radioactive pinion to the inactive gear was made by monitoring and radiographing the inactive gear after running at each load step. The results are shown in Fig. 5 in the form of radiographs of SAE 10w and SAE 60 straight mineral oils and SAE 10w compounded with an E.P. additive. Also included is one radiograph for gears operated without any lubricant at an extremely low speed, 100 rpm, for one minute with just enough load to insure positive contact.

The gear operated dry shows metal transfer all over the contacting surface, including the pitch line. In spite of the fact that the operating condition was exceptionally light, the metal transfer was quite noticeable by this technique and serves as a guide to compare the metal transfer of lubricated gears. The first appearance of metal transfer with SAE 10w oil occurred at 10 pounds load, and with SAE 60 at 20 pounds load. At these loads the first appearance of scoring at the dedendum of the pinion and the addendum of the gear was observed. In no case was metal transfer observed at the pitch line.

The SAE 10w oil compounded with an E.P. agent first showed metal transfer at 10 pounds load, just like the uncompounded SAE 10w oil; however, the amount was much less since scoring did not occur. The amount of metal transfer increased as the load steps increased, although no scoring took place. Again no metal transfer was detected at the pitch line.

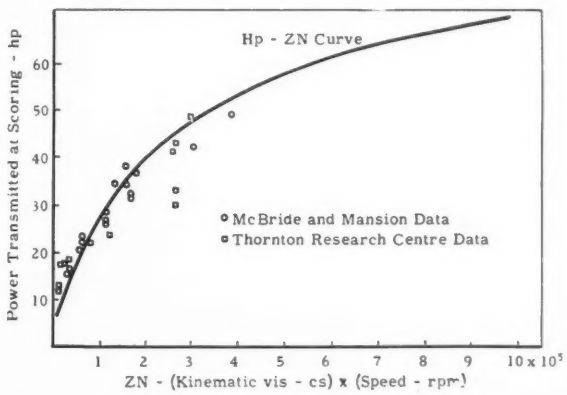


Fig. 6 Correlation of Emeryville Hp-ZN Curve with scoring data of other laboratories.

DISCUSSION: It is of interest to compare the results obtained by the authors with independently obtained data of other laboratories. Such data have been made available by the Thornton Research Centre of Shell Petroleum Co., and others are found in the published work of McBride and Mansion[10]. Both sets of data were obtained on IAE 3¼ inch gear machines operated at a constant oil temperature of 50 C. and 60 C., respectively. Uncompounded mineral oils were used, most of which contained less than 0.2% of sulfur. These data plotted as power transmitting capacity vs. the speed-viscosity product are shown in Fig. 6. The correlation with the previously shown Hp-ZN curve is excellent.

This Hp-ZN relation strongly suggests that the power transmitting capacity of a non-reactive mineral oil is directly dependent upon its rheological properties which must exert their influence through hydrodynamic pressure. The absence of metal transfer in tests with radioactive pinions and unreactive oils gives added support to the conclusion that thick film hydrodynamic conditions existed. Metal transfer through boundary films is theoretically possible and it has been shown by the work with an E. P. oil above the score load of its base oil, that such boundary films are detectable by the transfer technique.

The radioactive wear tests, much more sensitive than the previously used weighing techniques, lend further support to the presence of a thick film of mineral oils operating below their score loads. It must be admitted here, however, that with the test durations thus far employed the wear believed to be present when just the extremes of the gear tooth profile are operating under known boundary conditions, has not been detected.

The variable effect of temperature upon the

load carrying capacity of some oils as illustrated in Fig. 2 has been shown to be closely related to the presence of potentially reactive components, primarily sulfur. It seems logical to assume that the activity of such components in the boundary film forming reaction may increase with increasing temperature. Much more information is needed in this field.

Summarizing the above results indicate that the gears lubricated with an unreactive mineral oil operated in the thick film zone of lubrication as long as no scoring occurred. Gears lubricated with oils compounded with E. P. agents also operated in the thick film zone up to the score limits of their base oil, and loads above this limit were carried by an adsorbed film typical of the boundary zone of lubrication. Since operation in the thick film zone offers many advantages such as smaller friction and practical absence of wear, both gears and gear lubricants should be designed to operate in the hydrodynamic zone wherever possible. Where it is not possible to have thick film lubrication for the peak loads of gear operation, an E. P. film can serve to protect the gear against scoring at these peak loads with thick film lubrication supporting the normal operating loads. Only severe design limitations would seem to make continuous operation in the boundary zone seem acceptable.

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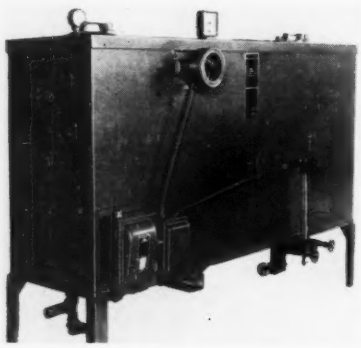
Lubrication in the News



WATER-OIL SEPARATOR: A new, redesigned Houdaille "Dri-Pure" water-oil separator is now offered by the Honan-Crane Corp., especially effective in the removal of moisture from vacuum pump oils thereby reducing pump-down time and increasing overall operating efficiency of vacuum pumps. Featuring two filtering compartments using "Palconia" (cellulose fibre) and "Cranite" (Fuller's earth) filtering media, plus a thermostatically controlled heating and settling tank, passage of the oil through these sections effects the oil filtration and water separation. Quick-opening lids over both filtering compartments offer access to handy, interchangeable filter cartridges, replaceable at intervals depending on the degree of oil contamination. (Heater unit may be connected to any convenient electrical outlet.) Easy to install and simple to operate, it can be hooked up to service one or more vacuum pumps depending on the application. Unit is also available for turbines, hydraulic systems and similar operations where water creates an oil contamination problem. For complete details, write: Honan-Crane Corp. (LE9/5), 818 Wabash Ave., Lebanon, Ind.

A LIST OF "DAG" DISPERSIONS, a pamphlet recently compiled by Acheson Colloids Co., lists all current "dag" dispersions, including their: dispersed substance, carrier, solids content, particle size, consistency, density (lbs./gal.), diluent, agitation before use, and typical application. For copies, write: Acheson Colloids Co. (LE9/5), 1630 Washington Ave., Port Huron, Mich.

GREASEMASTER UNIT: A newly developed battery-powered portable lubrication unit makes low-cost service-station-type grease jobs available at a fraction of the usual initial cost. An electrically driven hydraulic ram pump which is operated off an ordinary auto or tractor battery, the grease unit fits on top of any standard size 25#, 35#, or 40# grease container; a ten foot cable attaches to the battery and a twenty-five foot flexible greasing hose with a swivel coupler services all types of fittings. Pressure can be varied up to 12,000 lbs. per sq. in. for high pressure greasing, and a pressure regulator can be set to protect sealed bearings to prevent rupture. Complete specifications, photographs, and suggested applications are available by writing: G & T Industries (LE9/5), 1420 Barwise, Wichita, Kan.

TUTHILL PUMP CATALOGS: Two new catalogs featuring Tuthill small industrial pumps and general purpose pumps are now available: Catalog No. 101 covers the complete line of Model L Series of small industrial pumps designed for lubricating, hydraulic, transfer, circulating, and burning oils service. Catalog No. 102 presents the Model C Series of general purpose pumps for use with non-corrosive liquids with lubricating qualities. A unique feature of these catalogs is the pump guide in the front which presents the services, operating data, and distinguishing features for each model so that users can quickly determine the specific pump suited to their applications; dimensions and parts diagrams, performance curves, mounting styles, and modifications of pump

(Continued on page 271)

COOLING & LUBRICATION OF HIGH SPEED CYLINDRICAL ROLLER BEARINGS*

by W. J. Anderson & E. F. Macks
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Cleveland 11, Ohio

INTRODUCTION: Concurrent with the demands for more power and higher efficiencies in aircraft turbojet engines there has been a steady increase in the severity of operating conditions imposed upon the engine bearings. As a direct result of advances made in engine design a multiplicity of problems in bearing design and lubrication has arisen. The experimental information available on bearing performance at very high speeds has been lacking and far below the minimum required for intelligent design procedures.

The oil supplied to a high speed rolling contact bearing not only must lubricate but must also cool the bearing. At high speeds the oil flow required for lubrication is, in fact, small compared to that required for cooling. This is so because at high speeds bearing heat generation becomes significant and the oil must remove the major portion of this heat. In addition this heat must be removed from the oil to maintain safe operating temperatures for both bearings and oil. Finally it must be remembered that high speed flight imposes limitations on the availability of fluids to which heat can be rejected. It is therefore desirable to have the oil generate as little heat as possible while being churned and sheared within the bearing.

Information on the effects of oil viscosity and oil flow on operating temperatures, friction torque and power dissipated at low speeds is contained in Refs. 1, 2, and 3. In Ref. 4 the effects of oil temperature and viscosity on the load capacity and life of bearing materials is discussed. The effect of oil flow on friction torque and power dissipated has been studied to some extent in Refs. 5, 6, and 7.

The research to be discussed in this paper deals with the lubrication and cooling of a cylindrical roller bearing in which the heating of the bearing occurs only by the friction and churning within the bearing. The test conditions differ therefore, from actual service conditions where high engine ambient temperatures may cause heat flow to the bearing.

TEST EQUIPMENT & PROCEDURE: The bearing rig (Fig. 1) used for this investigation is described fully in Ref. 8. Briefly, the test bearing was mounted on one end of the test shaft which was supported in cantilever fashion so that component parts of the bearing and the lubricant flow could be observed during operation. Radial load was applied to the test bearing by means of a lever and deadweight system in such a manner that the outer race of the test bearing was essentially unaffected by small shaft deflections or by small shaft and load arm misalignments.

*Sponsored by the ASLE Technical Committee on Bearing Lubrication and presented at the ASLE 8th Annual Meeting, Boston, April 15, 1953.

The test bearing was a 75 mm. bore, 130 mm. outside diameter cylindrical roller bearing (size 215) of the type currently used as the turbine roller bearing in turbojet engines. Physical properties of the test bearing are given in Table I. The bearing was equipped with a one piece inner-race-riding bronze cage. Oil was supplied to the test bearing through a single jet of 0.050 inch orifice diameter directed at the space between the inner race and cage. In Ref. 9 this jet position was found to give the lowest average bearing temperature for this type bearing. Oil flow to the test bearing was varied from 2 to 9 lbs./min. by varying the oil inlet pressure from 20 to 400 lb./sq. in.

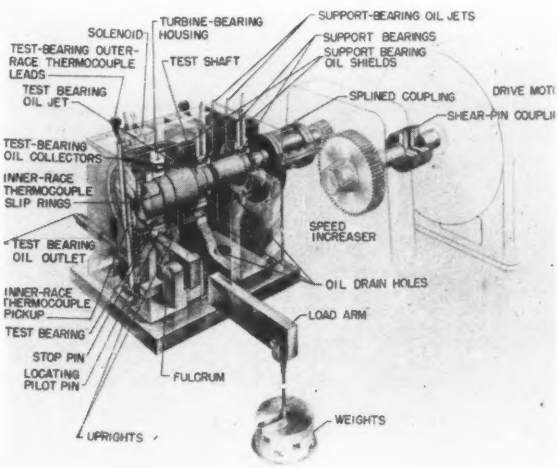

Fig. 1 Cutaway view of radical-load rig.

Five oils of widely different viscosities were used in the investigation. The viscometric properties of each of these oils are given in Fig. 2 and other properties are listed in Table II.

Although the data presented here were obtained from tests on just one bearing, past experience has shown that this type of data is highly reproducible. The test data is, therefore, considered valid and representative for any relatively new bearing of the type tested.

EXPERIMENTAL RESULTS & ANALYSIS. EFFECT OF DN: Fig. 3 shows the effect of DN value (product of bearing bore in mm. and shaft speed in rpm) on bearing outer-race temperature for each of the five test oils. The functional relationship between bearing temperature and DN is very nearly linear over the range of speeds investigated, and this relationship is not changed to any extent by varying the lubricant viscosity. It is seen that at a

specific DN value an increase in lubricant viscosity increases the bearing temperature.

In Fig. 4, it is seen that at a constant load and oil flow the temperature rise of both the deflected oil (that oil which merely splashes off the face of the bearing) and the transmitted oil (that oil which travels through the bearing) increases with an increase in DN. In effect this means that the power rejected to the oil increases with speed. The power rejected to the oil depends to a great extent on the oil flow rate and on the difference between the bearing operating temperature and the oil inlet temperature. As the surface speed increases the heat generated within the bearing and thus the bearing temperature tend to rise. The rate of heat transfer to the lubricant and to the surroundings increases until a balance is struck between the rates of heat generation and dissipation.

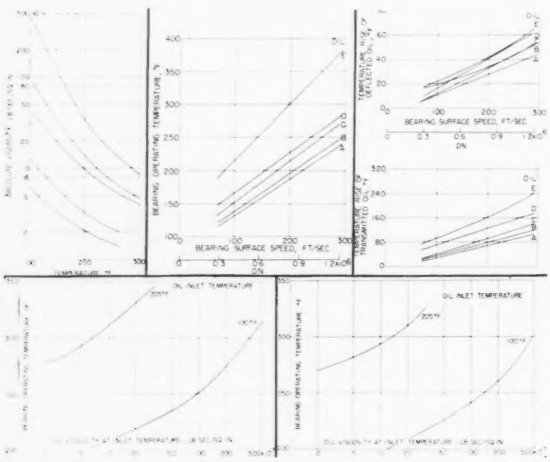

Fig. 2 (upper left) Absolute viscosities of five test oils.

Fig. 3 (upper middle) Effect of DN on bearing outer-race-maximum temperature for five oils. Load, 368 pounds; oil flow, 2.75 pounds per minute; oil inlet temperature, 100 F.; oil jet diameter, 0.050 inches.

Fig. 4 (upper right) Effect of DN on temperature rise of five oils. Load, 368 pounds; oil flow, 2.75 pounds per minute; oil jet diameter, 0.050 inches; oil inlet temperature, 100 F.

Fig. 5a (lower left) Effect of oil viscosity on bearing outer-race-maximum temperature. DN, 1.2×10^6 (surface speed, 290 feet per second); load, 368 pounds; oil flow, 5.9 pounds per minute; oil jet diameter, 0.050 inches.

Fig. 5b (lower right) Effect of oil viscosity on bearing inner-race temperature. DN, 1.2×10^6 (surface speed, 290 feet per second); load, 368 pounds; oil flow, 5.9 pounds per minute; oil jet diameter, 0.50 inches.

EFFECT OF OIL VISCOSITY: The effect of oil viscosity on bearing outer-race-maximum temperature is shown in Fig. 5 (a) and on bearing inner race temperature in Fig. 5 (b). Oil flow, load and speed are kept constant. Curves for two oil inlet temperatures are shown and indicate clearly the effect of oil inlet temperature on bearing temperature. Comparison of Figs. 5 (a) and 5 (b) reveals that, in the subject test rig, bearing outer-race-maximum temperature exceeds bearing inner race temperature and that the functional relationships between oil viscosity and both bearing outer-race-maximum and inner race temperatures are very similar.

Lubricant viscosity has a marked effect on bearing temperature. This suggests that the rate of heat generation within the bearing increases rapidly with increasing viscosity. There appear to be, however, two basic causes for the increase of bearing temperature with increasing lubricant viscosity: a source of heat within the bearing arising from the shearing and churning of the oil, and the decreased film coefficient of heat transfer between the bearing surfaces and the lubricant.

To determine qualitatively the effect of oil viscosity on churning losses use was made of a bearing heat balance. For equilibrium operation of the bearing the heat generated in the bearing will be equal to the heat rejected to the surroundings plus that absorbed by the oil so that (1):

$$H_{GB} + H_{GO} = H_{RS} + H_{RR} + H_{RD} + H_{RT}.$$

For a specific bearing environment (such as is obtained in a test rig) H_{RS} and H_{RR} will depend only on T_B and T_R and will be essentially constant for constant T_B and T_R. H_{GB} is assumed to be constant for a given bearing, load and DN value. With this assumption calculations can then be made to determine the effect of viscosity on H_{GO} without the necessity of measuring friction torque. Substitution of expressions for H_{RD} and H_{RT} in equation (1) yields (2):

$$H_{GB} + H_{GO} = H_{RS} + H_{RR} + M_D \int_{T_{OI}}^{T_{DO}} c\,dT + M_T \int_{T_{OI}}^{T_{TO}} c\,dT$$

Now for $T_B = 700$ F. absolute and $T_{OI} = 560$ F. absolute, the data shown in Table III were obtained. An expression for the specific heat of the oils (accurate to within 1 percent) is (3): $C = 0.195 + 0.000478\,T$, where T is in degrees Fahrenheit absolute. Substitution of equation (3) together with the data of Table III into equation (2) yields for each of the oils (4a): $H_{GB} + (H_{GO})_A = H_{RS} + H_{RR} + 61.2$, (4b): $H_{GB} + (H_{GO})_B = H_{RS} + H_{RR} + 82.2$, (4c): $H_{GB} + (H_{GO})_C = H_{RS} + H_{RR} + 96.2$, (4d): $H_{GB} + (H_{GO})_D = H_{RS} + H_{RR} + 116.4$.

From equations (4a) through (4d) the following relations are obtained (5a): $(H_{GO})_B - (H_{GO})_A = 21.0$ Btu/min $= 0.494$ hp, (5b): $(H_{GO})_C - (H_{GO})_A = 35.0$ Btu/min $= 0.823$ hp, (5c): $(H_{GO})_D - (H_{GO})_A = 55.2$ Btu/min $= 1.30$ hp.

The results of these calculations are plotted in Fig. 6. It is significant to note that, for the operating conditions investigated, the use of oil D results in a power loss of 1.30 hp greater than that for oil A due to churning.

The heat removed from the bearing by the deflected oil is given by the equation (6):

$$H_{RD} = M_D \int_{T_{OI}}^{T_{DO}} c\,dT$$

If, for purposes of approximation, the specific heat of the oil is assumed independent of temperature (this is valid in the following calculation because the average temperature of the five oils is very nearly the same) then (7): $H_{RD} = M_D\, C\, (T_{DO}-T_{OI})$.

In terms of a film coefficient of heat transfer, h, and an area of contact, A, between the oil and bearing (8): $H_{RD} = hA\,(T_B-T_{OI})$.

From equations (7) and (8), (9): $hA/C = M_D\,(T_{DO}-T_{OI})\,/\,(T_B-T_{OI})$.

Data at constant deflected oil flow, speed and load (shown in Table IV) were used to calculate

hA/C for the five test oils. The calculated values listed in the table show that the film coefficient of heat transfer for oil A is about twice that for oil E.

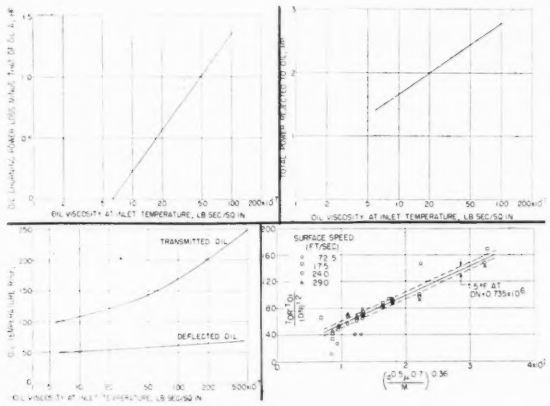

Fig. 6 (upper left) Effect of oil viscosity on power loss due to churning oil. DN, 1.2×10^6 (surface speed, 290 feet per second); load, 368 pounds; bearing outer-race-maximum temperature, 240 F.; oil jet diameter, 0.050 inches; oil inlet temperature, 100 F.

Fig. 7 (upper right) Effect of oil viscosity on total power rejected to oil. DN, 1.2×10^6 (surface speed, 290 feet per second); load, 368 pounds; bearing outer-race-maximum temperature, 240 F.; oil jet diameter, 0.050 inches.

Fig. 8 (lower left) Effect of oil inlet viscosity on oil temperature rise for a constant oil flow. DN, 1.2×10^6 (surface speed, 290 feet per second); load, 368 pounds; oil flow, 2.75 pounds per minute; oil jet diameter, 0.050 inches.

Fig. 9 (lower right) Test rig cooling correlation curve for outer-race temperature rise above oil inlet temperature. Surface speeds, 175 to 290 feet per second; oil inlet temperature, 100 to 205 F.; oil viscosity at inlet temperature, 2 to 470 x 10^{-7} reyns; load, 300 to 1100 pounds; oil flow 1 to 8 pounds per minute; jet diameter, 0.023 to 0.129 inches.

Inasmuch as it was not possible to measure friction torque with the subject test rig, the total power absorbed by the test bearing could not be computed. However, the total power rejected to the oil (the heat load on the oil system) could be computed. A typical curve of total power rejected to oil against oil inlet viscosity at a constant bearing temperature is plotted in Fig. 7. The operating conditions are identical to those which prevailed in Fig. 6. The total power rejected to the oil is roughly a linear function of the log of the oil viscosity at inlet temperature. Therefore, the higher the lubricant viscosity the greater must be the capacity of the lubricant cooling system to maintain a specific bearing temperature at a given load and speed. Since lubricant cooling constitutes a serious problem in present day jet engines, the use of higher viscosity lubricants will serve only to aggravate this problem.

Curves of oil temperature rise as influenced by oil inlet viscosity at a constant oil flow, load and speed are shown in Fig. 8. The curves of Fig. 8 are presented because they, together with the curves of Fig. 5, show conclusively that the power loss in a bearing of this type increases with increasing lubricant viscosity. To show the validity of the above statement it must be demonstrated that both the heat rejected to the surroundings and that rejected to the oil increase simultaneously with an increase in oil viscosity. Fig. 5 shows that bearing temperature increases with increasing oil viscosity;

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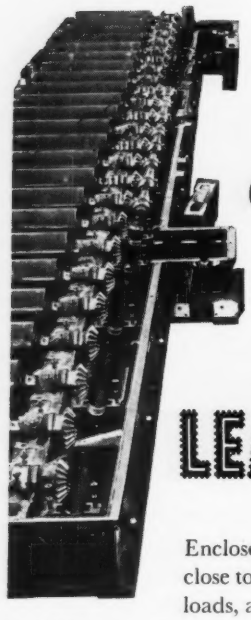

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this can only mean an increased rate of heat rejection to the surroundings. Likewise Fig. 8 shows that oil temperature rise increases with increasing oil viscosity at a constant oil flow and this can only mean an increased rate of heat rejection to the oil.

TABLE I

Bearing Type	Cylindrical roller	
Construction	One-piece inner-race-riding cage	
Number of rollers	18	
Roller length-diameter ratio	1	
Pitch diameter of bearing (in.)	4.036	
	Before	After
Total running time, (hr)	Unknown	+ 86.5
Severity factor [b]	Unknown	+ 6.26 X 10^5
Roller diameter, (in.)	[c] 0.5513	0.5500
Roller length, (in.)	[c] 0.5510	0.5500
Diametral clearance between cage and roller, (in.)	[c] 0.0087	0.006
Axial clearance between roller and inner-race flange, (in.)	[c] 0.002	0.003
Axial clearance between roller and cage, (in.)	[c] 0.006	0.008
Unmounted bearing:		
Diametral clearance, (in.)		
Bearing	0.0018	0.0015
Cage	.014	.0215
Eccentricity	.0001	.0000
Mounted bearing:		
Diametral clearance, (in.)		
Bearing	0.0004	0.0004
Cage	.013	.0205
Eccentricity	-------	-------

TABLE II

Oil	Pour Point (°F)	Flash Point (°F)	Viscosity Index	Autogenous Ignition Temperature, (°F) [a]
A	<-75	215	82	470
B	<-75	300	75	500
C	-50	310	150	490
D	-20	450	100	760
E	20	515	98	780

TABLE III

Oil	Deflected oil flow (lbs/min)	Deflected-oil outlet temperature (°F abs)	Transmitted oil flow (lbs/min)	Transmitted-oil outlet temperature (°F abs)
A	2.2	617	0.03	662
B	3.35	607.5	0.12	675
C	4.75	592.5	0.39	683
D	5.05	590	0.65	702

Table I (top) Physical characteristics of test bearing (obtained as in Ref. 7). Key: b—Summation of products of difference between equilibrium maximum bearing temperature and oil inlet temperature for each corresponding operating time in minutes at that particular condition. c—Measurements obtained from sample bearing.

Table II (middle) Properties of five test oils. Key: a—Determined in modified Moore type apparatus. Time lag before ignition at temperature indicated was under 2 minutes.

Table III (bottom) Data for calculation of oil churning losses.

EFFECT OF OIL INLET TEMPERATURE: Calculations were made to determine the effect of oil-inlet temperature on the power rejected to the oil at a constant bearing temperature. The data available made it possible to make just two calculations for oils B, C, D and E. The results of the calculations, shown in Table V, demonstrate that for a constant bearing temperature the total power rejected to the oil decreases slightly with increasing oil inlet temperature. The correct explanation for the above phenomenon, while unknown, may be either that the churning losses were lower at the higher oil inlet temperature or that the coefficients of heat transfer to the surroundings were influenced by oil inlet temperature.

COOLING-CORRELATION THEORY: A cooling correlation analysis, which resulted in the development of cooling-correlation curves for both inner- and outer-race temperatures of 75 mm. bore cylindrical roller bearings, was presented in Refs. 9, 10, and 11.

The basic equation developed was (10): $\Delta T/(DN)^a = B(\mu^b d^e/M)^f$, where B, a, b, e, and f are constants which may be determined empirically for a specific bearing and environment. The method of determining the exponents is given in Ref. 10. In the original presentation of the analysis, given in Ref. 10, data used to test the applicability of the analysis were obtained using only oil C of this report. The data presented herein in Fig. 9 were obtained using all five test oils. The correlation is thus found to be fairly good over the range of viscosities from 2×10^{-7} to 470×10^{-7} reyns at inlet

temperatures. Although only the outer-race cooling-correlation curve is presented herein, similar results were obtained with the inner-race correlation.

For the conditions investigated the slope of the outer-race cooling-correlation curve was found to be 4.3×10^{-4}. This is within 2.5 percent of the value of 4.42×10^{-4} obtained in Ref. 10.

The estimated range of applicability of the outer-race cooling-correlation curve (Fig. 9) is as follows: DN 0.735×10^{6} to 1.2×10^{6}; oil viscosity at inlet temperatures, 2×10^{-7} to 470×10^{-7} reyns; oil flow, 2 to 10 pounds per minute; oil-inlet temperature, 100 to 205 F.; oil-jet diameter, 0.023 to 0.129 inches, and load, 300 to 1100 pounds. Points for a DN value of 0.3×10^{6} are shown but the degree of correlation is poor.

TABLE IV

Oil	W_D (lb/min)	T_B (°F abs)	T_{DO} (°F abs)	hA/C (calculated)
A	3	686	606	1.095
B	3	706	612	1.068
C	3	722	614	1.009
D	3	735	608	0.822
E	3	828	611	0.57

TABLE V

Oil	Oil inlet temperature (°F abs)	Deflected oil temperature maximum bearing-race temperature (°F abs)	Bearing load (lb)	Deflected oil flow (lb/min)	Transmitted oil flow (lb/min)	Deflected oil outlet temperature (°F abs)	Transmitted oil outlet temperature (°F abs)	Power rejected (hp) To deflected oil	Power rejected (hp) To transmitted oil	Power rejected (hp) Total
B	560	732	1.8	0.01	633	685		1.47	0.01	1.48
	665		8.6	0.87	673	718		0.85	0.57	1.42
C	560	752	1.8	0.01	658	721		2.03	40.01	2.03
	665		8.3	0.75	673	735		0.87	0.66	1.53
D	560	760	1.9	0.07	634	740		1.59	0.15	1.74
	665		7.2	0.65	676	750		0.97	0.70	1.67
E	560	800	1.9	0.14	594	775		1.52	0.36	1.81
	665		7.5	0.45	689	784		0.76	0.67	1.43

Table IV (top) Data for calculation of film coefficients of heat transfer.

Table V (bottom) Power rejected to oil for two oil inlet temperatures.

SUMMARY: A 75 mm. bore cylindrical roller bearing was used in an experimental investigation to determine the effects of oil viscosity on bearing operating characteristics over a range of DN values from 0.3×10^{6} to 1.2×10^{6}. The cooling correlation analysis of Ref. 10 was verified for a wide range of oil viscosities. A similar cooling correlation curve could be developed for any bearing installation to enable designers to predict bearing temperature rise from a single curve regardless of whether bearing speed, bearing load, oil flow, oil inlet temperature, oil inlet viscosity, oil inlet velocity, or any combination of these parameters vary over wide ranges.

Minimum bearing temperatures resulted (with load, speed, oil flow and oil inlet temperature constant) when a low viscosity, low viscosity-index oil was used.

Minimum power rejection to the oil (at a constant bearing temperature) resulted when a low viscosity oil was used introduced at high inlet temperatures.

In the viscosity range investigated bearing temperatures increased with increasing oil viscosity when speed, load and oil flow were kept constant. A higher oil flow of a more viscous oil was therefore required to maintain a constant bearing temperature.

NOMENCLATURE

A heat transfer area, sq in.
C specific heat of oil, Btu/lb °F
D diameter of bearing bore, mm
d oil jet diameter, in.
h film coefficient of heat transfer, $\frac{\text{Btu}}{(\text{min})\ (\text{sq in.})\ (°\text{F})}$
H_{GB} heat generated due to bearing friction, Btu/min
H_{GO} heat generated in churning oil, Btu/min

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H_{RS} heat transferred to the surroundings by conduction and convection, Btu/min
H_{RR} heat transferred to the surroundings by radiation, Btu/min
H_{RD} heat removed by the deflected oil, Btu/min
H_{RT} heat removed by the transmitted oil, Btu/min
M mass flow of oil, lb/min
M_D deflected oil flow, lb/min
M_T transmitted oil flow, lb/min
N bearing speed, rpm
T_B bearing temperature, °F abs
T_{DO} deflected oil outlet temperature, °F abs
T_{IR} bearing inner race temperature, °F
T_{OI} oil inlet temperature, °F abs
T_{OR} bearing outer race temperature, °F
T_{TO} transmitted oil outlet temperature, °F abs
μ absolute viscosity of oil, $\frac{\text{sq in.}}{\text{lb sec}}$

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(LUBE LINES, Continued from page 245)

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Book Reviews

PRESSURE-VISCOSITY REPORT (2 Volumes)
by ASME, Research Dept., 29 W. 39th St., New York 18, N. Y., 1953, price $10.00.

Recent experimental work has resulted in comprehensive data on the viscosity and density of more than forty lubricating fluids at temperatures from 32 to 425 F., and at pressures up to 150,000 psi; these new data are presented in PRESSURE-VISCOSITY REPORT issued by the Research Committee on Lubrication of the American Society of Mechanical Engineers. An Advisory Board composed of experts from universities, government laboratories, and the petroleum, synthetic lubricant and equipment building industries contributed their services to insure representative selection of test fluids and to advise on the conduct of the experimental work; petroleum and government laboratories made physical and chemical inspection tests on the fluids to insure that all fluids tested were properly identified; the pressure-temperature-viscosity-density measurements were carried out at Harvard University under the direction of Prof. R. V. Kleinschmidt (This data has been entered on IBM punch cards and sets may be obtained from the ASME Research Dept. at a nominal charge.). Data presented in this report will be subjected to many analyses in connection with lubrication problems; methods of expressing the variation of viscosity and density with pressure and temperature and in correlation of these properties with the com-

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position of lubricating fluids. (Small quantities of many of the fluids have been stored in sealed ampules for future reference or testing if such is needed.)

APPROVED LUBRICATION GUIDE FOR TRUCKS
(5th Edition)
by Chek-Chart Corp., 31 E. Congress Parkway, Chicago 5, Ill., 1953, 109 pages, unit price includes one-year subscription to Chek-Chart Service Bulletin.

Correct service for the nation's trucks, according to the recommendations of the truck manufacturers, is the keynote of APPROVED LUBRICATION GUIDE FOR TRUCKS. Covering the majority of trucks in use in the United States, 55 truck diagram pages and 18 tabular pages on heavy trucks present complete service and lubrication data; also included are 6 pages of step-by-step service instructions, a composite lubrication chart for truck cabs, and 12 diagram charts of typical axle and trailing axle units. The new edition presents tabular listings of crankcase, transmission and differential capacities and lubricant recommendations for busses and motor coaches.

Personals

A. C. Withrow Co., Los Angeles, developers of special industrial lubricants, has announced the appointment of H. L. FARROW as Plant Manager.

J. A. BRADY, Detroit, has been appointed Manager of the Lubrication Dept. in charge of Sales & Application of Products, Penn-Petroleum Corp.

The Dingle-Clark Co., contracting engineers, has announced the transfer of R. V. ROBISON to Manager of the Machinery Div. of their Cleveland territory, with L. B. ABRAMS, JR., replacing Mr. Robison as Sales Engineer in their Pittsburgh office.

A. W. LINDERT has been appointed an Assistant Division Director, a newly-created post, in the Whiting, Ind., Research Labs. of the Standard Oil Co.

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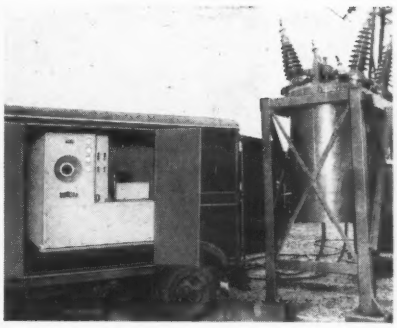

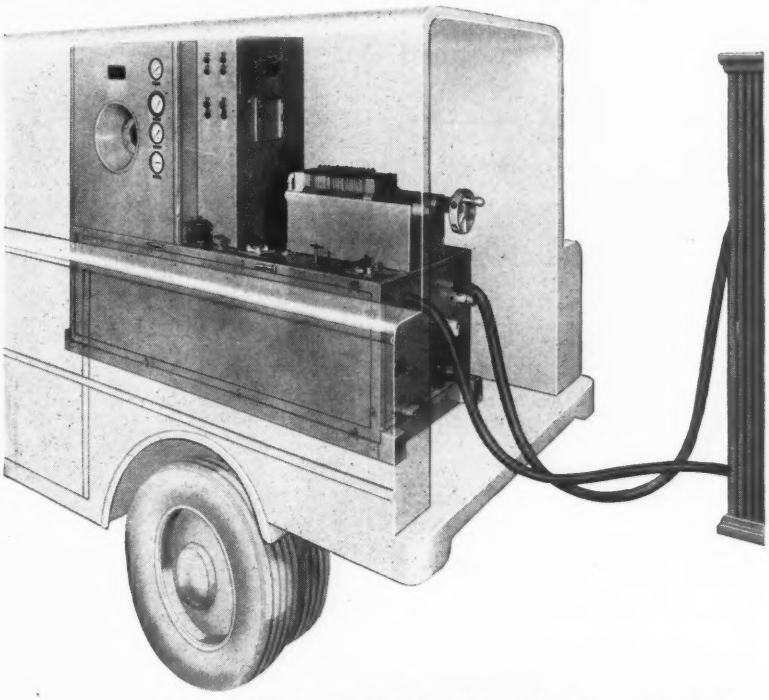

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C. O. MARTIN, Attapulgus Minerals & Chemicals Corp., Phila., has been assigned to sales and technical service activities in connection with Permagel, the company's new thickening agent for organic and aqueous media.

Commercial Filters Corp. has announced the opening of a Cincinnati Sales Office at 6908 Silverton Ave., under the direction of R. P. FINDLOW, formerly of their New York office.

R. E. PETERSON, Westinghouse Research Labs., East Pittsburgh, was presented an Award of Merit for outstanding service particularly along technical lines in ASTM committees, and J. P. HAMER, Standard Oil Development Co., co-authored a paper entitled "Lubricating Oil Requirements of the Modern Automotive Engine," at the ASTM Annual Meeting held in Atlantic City.

The E. W. Dumler Co. announces the appointment of H. R. DRYLIE as Sales Manager of their new Pittsburgh Office, 6324 Penn Ave., where a complete line of automotive-industrial filters and replacements will be stocked.

Two sales departments of the Monsanto Chemical Co., Organic Chemicals Div., St. Louis, have been combined into the Dept. of Petroleum Chemicals & Functional Fluids Sales, with J. W. NEWCOMBE named as Manager.

The National Conference on Industrial Hydraulics is sponsoring an evening course (Wednesdays, Sept. 23 throughout the Fall semester) entitled "Introduction to Oil Hydraulic Power," at the Illinois Institute of Technology Center, 3300 S. Federal St., Chicago; lecturers will include J. J. SLOMER, Goodman Mfg. Co., and J. C. VanGUNDY, The Texas Co.

(Lube in the News, from 262)

design are included. For copies, write: Tuthill Pump Co. (LE9/5), 939 E. 95th St., Chicago 19, Ill.

OPEN GEAR & CABLE LUBRICANT: Tenac, a free-flowing, fast setting, highly adhesive lubricant for open gears and cables, is described in a new 4-page bulletin by E. F. Houghton & Co. Ease of application by spray, brush, or swab, and range of application are outlined along with a dozen distinct advantages. Product is described as a black lubricant having an extremely viscous petroleum base fortified with special additives to provide exceptionally high adhesiveness and extreme pressure characteristics; folder also gives information on grades available. For copies, write: E. F. Houghton & Co. (LE9/5), 303 W. Lehigh Ave., Phila. 33, Pa.

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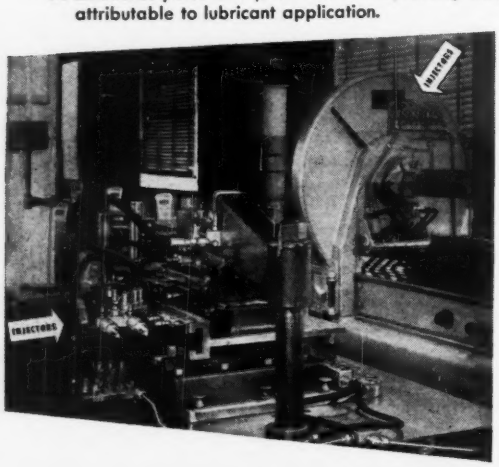

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PIPE REPAIR CLAMP: Details of the method by which pipe line leaks of any kind can be permanently repaired in but a matter of seconds, is covered in free literature now available on Morris band-type Repair Clamps. Folder No. 201 illustrates method of installation for both single and double repair clamps; covers complete ordering specifications and prices. For copies, write: Morris Coupling & Clamp Co. (LE9/5), Dept. M-19, Ellwood City, Pa.

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systems, strengthener and suspending agent. Permagel will be available in carlots by the first of the year—complete technical data and generous samples available now by writing: Attapulgus Minerals & Chemicals Corp. (LE9/5), 210 W. Washington Sq., Phila. 5, Pa.

HOT BOX TESTING MACHINES: Completion of two unusual testing machines for use in research on minimizing the occurrences of railroad freight car hot boxes has been announced by the Armour Research Foundation of the Illinois Institute of Technology. The testing machines are: (1) An actual freight car truck—the assembly containing four wheels at the front or rear of a freight car—is set up so that cooling, heat flow, and temperature characteristics of the journal can be studied under controlled laboratory conditions. Foundation scientists plan to study how the bearing is cooled and how the cooling action may be improved. (2) A full-scale bearing testing machine, which simulates the workings inside a journal box under controlled laboratory conditions. The tester will be used in measuring frictional resistance of the bearing; fluid pressures; effects of different oils; and characteristics of waste materials used in lubrication.

Patent Abstracts

Prepared by ANN BURCHICK from OFFICIAL GAZETTE, Vol. 671, No. 5; Vol. 672, Nos. 1, 2, 3, 4; Vol. 673, Nos. 1, 2, 3. Printed copies of patents are available from the Patent Office at twenty-five cents each. Address the Commissioner of Patents, Washington, D. C. for copies and for general information concerning patents.

TREATMENT OF ELECTRIC APPARATUS INTENDED TO CONTAIN OIL, Patent #2,643,962
by J.H.O. Lindhe, assignor to A. Johnson & Co.

A process of preventing sludge formation and oxidation of the oil in transformers, which comprises treating the metal parts of a transformer, which come into contact with the transformer oil, with a mineral oil containing dissolved therein a small amount of a metal dithiophosphate, selected from a class consisting of barium, zinc and calcium dithiophosphates, under conditions causing the formation of a phosphorus-containing compound on the metal parts, removing the dithiophosphate-containing oil and replacing it with transformer oil free from dithiophosphates.

NONSTICKING WAX ARTICLE AND METHOD OF PREPARATION, Patent #2,643,973
by N. Thompson, assignor to Sun Oil Co.

An article of manufacture comprising a body of petroleum wax carrying a surface film, in amount effective to prevent sticking, of hydrocarbon distillate oil having a viscosity of 40-7000 S.U. seconds at 100 F.

DEMULSIFIABLE LUBRICANT COMPOSITIONS, Patent #2,643,979
by A. W. Lindert, assignor to Standard Oil Co.

A lubricant composition comprising essentially a major proportion of a hydrocarbon oil of from about 0.5% to about 15% of an organic extreme pressure agent containing an element selected from the group consisting of chlorine, sulfur, phosphorus, and mixtures thereof, and normally susceptible to the formation of oil-in-water emulsions in the presence of water, from 0% to about 10% of an asphalt, and from about 0.005% to about 1% of a compound selected from the group consisting of a quaternary compound.

COMPOUNDED LUBRICATING OIL, Patent #2,644,792
by M. W. Hill, assignor to Standard Oil Development Co.

A mineral oil containing a detergent quantity of a product obtained by reacting one molecular proportion of a phosphorus sulfide with 1 to 10 molecular

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proportions of a hydrocarbon at about 200 to about 600 F. and treating the acidic reaction product at about 100 to about 400 F. with an amount sufficient to neutralize the titratable acidity of such reaction product of an organic basic reaction compound.

RUST INHIBITING COMPOSITION, Patent #2,644,793
by H. W. Rudel & Marion Gargisa, assignors to Standard Oil Development Co.

A mineral lubricating oil containing dissolved therein 0.001 to 5% of the compound of the formula $RCOOCH_2CH_2SCH_2COOH$ where R is a hydrocarbon radical selected from the group consisting of aliphatic and cycloaliphatic radicals and containing 2 to 21 carbon atoms.

METAL WORKING LUBRICANT, Patent #2,645,614
by W. C. Holmes, assignor to Tap and Drill Ez Corp.

A tool lubricant composition in a paste form consisting essentially of substantially two parts paraffin wax, three parts beeswax, one part machine lubricating oil, and one part carbon tetrachloride on a weight basis, with a portion of the carbon tetrachloride disposed through the paste in the form of droplets.

LIGHT DISTILLATE COMPOSITIONS, Patent #2,646,349
by D. A. Wagner, assignor to Sinclair Refining Co.

A composition of matter comprising predominantly a light oil product and containing 1 to 50 pounds of sulfurized tall oil per 1,000 barrels of the light oil produced.

PETROLEUM HERBICIDE EMULSIONS, Patent #2,646,350
by F. S. Rostler, assignor to Golden Bear Oil Co.

As a herbicidal composition of matter, an emulsion comprising water as the continuous phase and as dispersed phase, a crude oil component, said component being a resinous petroleum fraction having a flash point above 250 C. substantially free of asphaltenes, containing saturated components in amount not greater than about 25% of the resinous fraction, and having a gravity of at least one, and a herbicide dissolved in said resinous petroleum fraction.

LUBRICANT & METHOD OF MAKING SAME, Patent #2,637,722
by D. Frazier, assignor to The Standard Oil Co.

A method of improving the color of lubricants and lubricant additives which are reaction products of 5 to 60% by weight of a phosphorus sulfide and 95 to 40% of a mixture of one part by weight of an oxygen-containing organic compound selected from the group consisting of those compounds containing hydroxy, carbonyl, ether and carboxy radicals having a boiling point of at least 225 F. with 1 to 10 parts of a polyolefin having a molecular weight of about 2000 to 100,000 and capable of improving the viscosity index characteristic of a lubricating oil, reacted together at a temperature in the range of 225 to 500 F., which method comprises treating such reaction products with an amount of an alkali hydroxide in the range of 0.1 to 0.75% by weight of the reaction mixture and sufficient to improve the color of the

reaction product and then treating the resulting reaction product with an amount of a hydrogen peroxide compound in the range of 0.15 to 0.5% by weight of the reaction mixture and sufficient to improve the stability of the reaction product and recovering the improved reaction product.

LUBRICANT FOR PNEUMATIC TOOLS & THE LIKE, Patent #2,638,446
by J. I. Wasson, assignor to Standard Oil Development Co.

A lubricant for pneumatic tools, and the like, consisting essentially of about 86 to 97 parts by weight of mineral base lubricating oil having a viscosity between 50 and 500 SSU at 100 F., 1 to 3 parts of degras, 0.1 to 2 parts of oil-soluble calcium petroleum sulfonate, 1 to 5 parts of sulfurized sperm oil containing 10 to 20% of sulfur, 0.5 to 2 parts of tricresyl phosphate and 0.5 to 2 parts of polybutene of molecular weight between 10,000 and 50,000.

GEAR LUBRICANT, Patent #2,638,447
by F. P. Otto, L. W. Manley & R. V. White, assignors to Socony-Vacuum Oil Co., Inc.

An extreme pressure lubricating composition comprising a major proportion of a mineral lubricating oil and minor proportions of addition agents as follows: (a) from about 1% to about 10% of a product obtained by chemically substituting a part only of the chlorine in a chlorinated aliphatic hydrocarbon material having from 2 to about 24 carbon atoms with a thiocarbonate group, (b) from about 0.1% to about 3% of dihydroabietyl malate and (c) from about 0.1% to about 1.5% of the reaction product formed by reacting 3 moles of an alkylated phenol with about 1 mole of phosphorus pentoxide, at a temperature of from about 75 C. to about 125 C.

MINERAL OIL, Patent #2,638,448
by J. J. Wehrle, assignor to Socony-Vacuum Oil Co., Inc.

A mineral lubricating oil containing an amount sufficient to improve the oxidation characteristics thereof of the reaction product obtained by reacting a monohydric reactant selected from the group consisting of alkanethiols having between about 6 and about 24 carbon atoms per molecule and aliphatic alcohols having between about 6 and about 24 carbon atoms per molecule, and a phosphorus halide, in a molar proportion varying between about 0.5:1, respectively, and about 2,5:1, respectively, at a temperature of between about 0 C. and about 150 C., and for between about 3 hours and about 10 hours, to produce an intermediate product, and reacting a thiophenethiol with said intermediate product, in a molar proportion of said thiophenethiol to said phosphorus halide used to produce said intermediate product varying between about 2.5:1, respectively, and about 0.5:1, respectively, at a temperature of between about 0 C. and about 150 C., and for between about 3 hours and about 10 hours; the sum of the number of moles of said monohydric reactant and of said thiophenethiol so reacted with each mole of said phosphorus halide being three moles.

ANTICLOGGING FUEL OIL COMPOSITIONS, Patent #2,639,227
by E. B. Glendenning, C. Wies & J. B. R. Caron, assignors to Shell Development Co.

A non-clogging fuel oil normally susceptible to cause clogging due to the presence of cracked components in the fuel oil, containing in combination therewith from 0.0025% to about 0.05% of an oil-soluble calcium petroleum sulfonate and from 0.0015 to 0.0042% of oil-soluble calicum salt of octylphenol-formaldehyde condensation product.

LUBRICATING GREASE COMPRISING A COMPLEX ESTER BASE & SODIUM MYRISTATE, Patent #2,639,266
by J. P. Dilworth & C. H. Culnane, assignors to The Texas Co.

A lubricating grease consisting essentially of an oleaginous liquid lubricating base, about 15-25% by weight based on the grease of a sodium soap-forming fatty material containing at least 70% based on the fatty material of myristic acid 0.5-5% by weight of N, N' diphenyl p-phenylene diamine, 1.5-15% by weight of tricresyl phosphate and 0.1-1.0% of free alkali, said oleaginous liquid lubricating base comprising in minor proportion a mineral lubricating oil and in major proportion a complex polyester of lubricating characteristics obtained by reacting together under esterification conditions a polypropylene glycol containing from 6 to 9 carbon atoms, sebacic acid and 2-ethyl hexanol.

ADDITIVES FOR USE IN EXTREME PRESSURE GEAR OILS, Patent #2,640,030
by F. B. Fischl & M. W. Hill, assignors to Standard Oil Development Co.

An extreme pressure additive composition for lubricants, consisting essentially of a mineral lubricating oil base stock of 50 to 250 SSU viscosity of 210 F., treated with a phosphorus sulfide at a temperature within the range of 375 to 475 F. for a period of time of about 1-20 hours to incorporate about 3-6% by weight of phosphorus and about 5-12% of sulfur into the oil, said treated oil being stabilized by further treatment with 5 to 30% of its weight of a material selected from the class which consists of sulfurized fatty oils and their lower alcohol esters at a temperature in the range of 250 to 450 F., for 2 to 10 hours.

COMPOUND LUBRICATING OIL, Patent #2,640,053
by M. W. Hill & R. H. Jones, assignors to Standard Oil Development Co.

As a new composition of matter, a product obtained by reacting about one molecular proportion of a phosphorus sulfide with two to five molecular proportions of a hydrocarbon and further reacting the acidic product thus formed with 0.1% to 50% of its weight of a hydrocarbon containing at least one olefinic double bond.

DEMULSIFIABLE LUBRICANT COMPOSITIONS, Patent #2,640,811
by H. Sellei Beretvas, assignor to Standard Oil Co.

A lubricant composition resistant to the formation of stable emulsions with water comprising essentially a major proportion of a hydrocarbon lubricating oil and

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the following constituents in the following approximate weight percentages:

	Per cent
Oil-soluble lead soap	2 to 20
Organic sulfur-containing EP agent	3 to 20
High molecular weight fatty acid of at least 12 carbon atoms	0 to 10
Asphalt	1 to 25
Solvent extract of hydrocarbon oil	0 to 25
Oil-soluble alkaline earth sulfonate	0.25 to 3
Sulfated castor oil	0.25 to 2

said organic sulfur-containing E. P. agent being selected from the group consisting of a sulfurized fat, a sulfurized fatty acid, a sulfurized ester, a sulfurized vegetable oil and a sulfurized olefin, and said solvent extract of hydrocarbon oil containing compounds selected from the group consisting of aromatic compounds, polynuclear naphthenic compounds, and mixtures thereof.

GREASE, Patent #2,640,812
by W. C. Bryant, assignor to Swan-Finch Oil Corp.

A lubricating composition of a petroleum lubricant, a montmorillonite modified by an organic cation, and a long-chain organic salt gelator which contains a straight chain of at least 8 carbon atoms and whose salt-forming metal is selected from the group consisting of lithium, sodium, calcium, barium, and strontium, said organic salt having the characteristic of improving the lubricity and inhibiting the rust-producing properties of the modified montmorillonite the proportions of modified montmorillonite to said organic salt gelator ranging from 19-78% of said modified montmorillonite to 81-22% of said organic salt gelator.

LITHIUM - CALCIUM LUBRICATING GREASE COMPOSITION, Patent #2,641,577
by R. O'Halloran, assignor to Standard Oil Development Co.

A lubricating composition which comprises a lubricating oil thickened to a grease consistency with a mixture of the lithium and calcium soaps of an unsubstituted substantially saturated high molecular weight saponifiable material wherein the molar ratio of the lithium soap to the calcium soap is within a range of from 6:1 to 3:1.

COMPOUNDED LUBRICATING OIL, Patent #22,641,578
by C. A. Weisel, assignor to Standard Oil Development Co.

A composition consisting essentially of a mineral lubricating oil containing dissolved therein 0.01 to 10% of an amine salt of pentachlorophenol, the amine portion of said salt containing a total of not less than eight carbon atoms and is a member of the class consisting of aliphatic and cycloaliphatic amines.

SILOXANE BOUNDARY LUBRRICANTS, Patent #2,642,395
by C. C. Currie, assignor to Dow Corning Corp.

A composition of matter consisting essentially of (1) a liquid copolymeric siloxane having a degree of substitution of greater than 2 and less than 3 organic radicals per silicon atom, said siloxane containing halogenated aryl radicals selected from the group consisting of halogenated phenyl radicals and halogenated xenyl radicals, said radicals having substituted thereon from 1 to 7, inclusive, halogen atoms selected from the group consisting of chlorine and bromine, the remaining organic radicals in said siloxane selected from the group consisting of methyl and phenyl radicals, in which siloxane the halogenated aryl radicals are present in amount so that the ratio of halogen atoms to silicon atoms is from .05 to 5 and in which siloxane at least 27 per cent of the total number of organic radicals are methyl radicals and (2) from .01 to 5 per cent by weight based on (1) of a metal selected from the group consisting of copper, magnesium, and lead, said metal being present in the form of the salt of an acid selected from the group consisting of aliphatic carboxy acids having 4 to 9 carbon atoms, inclusive, and cyclohexyl derivatives of such aliphatic acids.

HYDROCARBON OIL CONTAINING A MERCAPTOBENZIMIDAZOLE, Patent #2,642,396
by T. C. Roddy, Jr., assignor to The Texas Co.

A mineral hydrocarbon oil normally corrosive to metal in the presence of sulfur containing a corrosion inhibiting amount within the range of 0.1 to 60 pounds per thousand barrels of oil of a corrosion inhibitor selected from the group consisting of mercaptobenzimidazole and alkyl substituted derivatives thereof.

ACYLATED POLYSTYRENE, Patent #2,642,398
by J. M. Butler, assignor to Monsanto Chemical Co.

A mineral oil containing dissolved therein 0.1% to 10% by weight of polystyrene of average molecular weight between 3,000 and 1,000,000 and which has been acylated with a monocarboxylic fatty acid chloride containing from 8 to 20 carbon atoms, which acid chloride is reacted to the extent of 0.10 to 1.0 mole of said chloride to each of the recurring units of the said polymer.

SYNTHETIC LUBRICATING OIL, Patent #2,642,452.
by L. A. Mikeska & P. V. Smith, Jr.,

Aesop's Fable of

"The Eagle and the Crow"

carries a moral for Cutting Fluid users

The Fable . . .

A Crow, sitting on a fence post, watched a huge Eagle swoop down on a luckless lamb and carry it away. "That looks easy," thought the Crow, and straightaway tried to duplicate the Eagle's feat. Of course, he couldn't lift the lamb and, instead, got tangled up in its wool and couldn't get loose. The farmer came over, grabbed the Crow, and wrung its foolish neck.

The Moral . . . Just as there was a big difference in abilities of the Eagle and the Crow in the fable, so are there big differences in the abilities of different types of cutting fluids. Stuart Oil Co. does not attempt to apply a watermixed cutting fluid where a straight oil is needed, or vice versa. Some applications require a fluid with great *lubricity*, or high *anti-weld* properties. All require *temperature regulation*, but always *more than a "coolant" is needed.*

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More Than a "Coolant" is Needed

assignors to Standard Oil Development Co.

A synthetic lubricating oil having an ASTM pour point below about -35 F., a viscosity index above about 125, a viscosity at 210 F. of at least 3.0 centistokes and a flash point of at least 450 F., said oil consisting predominantly of C_8 to C_{14} highly branched trialkyl esters of phosphoric acid, the ester portions of said phosphoric acid esters being derived from a primary monohydric alcohol boiling below about 450 F. which is obtained by Oxo synthesis wherein there is employed as the olefinic substance a hydrocarbon material whose olefin content consists of not more than 20% by weight based on the total olefin of an olefin of the type $RHC{=}{=}CH_2$ where R is a straight chain saturated aliphatic hydrocarbon radical.

PHOSPHORUS - CONTAINING LUBRICATING OIL ADDITIVES, Patent #2,643,261
by A. H. Matuszak & C. A. Weisel, assignors to Standard Oil Development Co.

As a new composition of matter, a phosphorus-containing ester prepared by reacting together pentaerythritol monooleate and a C_1 to C_{30} monohydroxy aliphatic alcohol at temperatures not above reflux with a phosphorus-containing material selected from the group which consists of $POCl_3$, PCl_5, and $PSCl_3$, in the presence of pyridine and a solvent, said ester being stripped of water and of said pyridine and solvent after reaction is substantially complete.

Current Literature

DETERMINATION OF HYDROPEROXIDES IN PETROLEUM PRODUCTS
by D. C. Walker & H. S. Conway, Analytical Chemistry, Vol. 25, No. 6, June 1953, pp. 923-925.

In this method for the determination of hydroperoxides in petroleum products, aqueous sodium arsenite is used as the reducing agent in a two phase system, and organic materials are removed by separation and extraction before determination of excess arsenite. Method is more sensitive than the iodometric method, and avoids the dependence on sample size characteristic of the ferrous method. Known concentrations of hydroperoxides in gasolines, fuel oils, and white mineral oils are determined with an accuracy of 0.03 hydroperoxide number (milliequivalents of active oxygen per liter of sample) for numbers less than 1, and within 2% for higher hydroperoxide numbers.

DEVELOPMENT OF A HIGH-PRESSURE VISCOSIMETER, THE
by M. L. Kimmel, Producers Monthly, Vol. 17, No. 6, April 1953, p. 6.

A rotating cylinder type viscosimeter has been developed and tested at atmospheric conditions for the ordinary range of reservoir crude oil viscosities of one-half to thirty centipoises. Additional informa-

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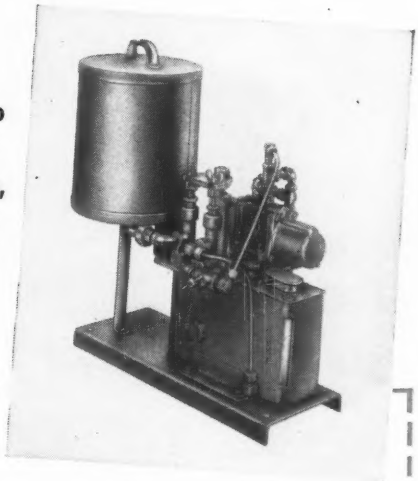

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tion from a variety of sources has been included to aid the reader in understanding the importance of viscosity in reservoir problems and the need of a new viscosimeter. The theoretical basis has been outlined for the torque reactions in this magnetic-drive, rotational viscosimeter.

EMULSIONS & STRAIGHT CUTTING FLUIDS

by A. E. Lawson & E. G. Ellis, Scientific Lubrication, Vol. 5, No. 6, June 1953, pp. 22-24.

Although use of an emulsion or a straight cutting oil will be conditioned by material to be machined, the particular tool, or the requirement of the operation itself, these two physically different fluids will do the same work. Reason for this is that in metal cutting, the high pressures and surface conditions call for active surface qualities in the fluid much more than the bulk properties such as density and viscosity. Cutting fluid requirements, the active lubricants in both fluids, influence of viscosity, and emulsion instability are taken up.

FOR MANUFACTURE OF LITHIUM GREASE

by P. J. Baker & N. T. Joyner, Institute Spokesman, Vol. 17, No. 3, June 1953, pp. 18-20, 22, 24.

For the past ten years the Girdler Corp. has been engaged in the development of continuous grease manufacturing processes. An apparatus has been developed for rapid, continuous multi-purpose type (lithium) grease production. The effect of temperature, type of oils and manufacturing methods on the properties of the finished greases are revealed. In this process the soap is prepared in the conventional way, using approximately one-half of the total oil required for the batch. The kettle temperature is adjusted to obtain a blend temperature in the critical range between 230 F., and 310 F.

GREASE LUBRICANT FOR DRILLING RIGS & TOOLS

by Anon, Petroleum Equipment, Vol 16, No. 2, April-May-June 1953, p. 84.

A recently developed grease lubricant for drilling rigs and tools is reported to possess properties which permit its use on a wide variety of grease type applications operating under favorable or unfavorable conditions. In addition, to its very high film strength, and adhesiveness, it resists the action of steam and water to a remarkable degree, and is becoming well known among drillers as a general purpose lubricant for all oil well drilling equipment—rigs, tools, drill joints, etc.

HYDRAULIC POWER & THE INDUSTRIAL TRUCK

by Anon, Lubrication, Vol. 39, No. 6, June 1953, pp. 77-88.

Successful mechanized materials handling depends on unit load and the industrial truck. The types of industrial trucks most commonly employed and their prime functions are described, and fundamentals of hydraulic systems and their application to industrial trucks are taken up. 70% of troubles in hydraulic systems are due to improper condition of hydraulic oil, perhaps the largest single factor being oxidation. Even small amounts of sludge can cause the closely fitted parts in the pump to stick and the orifices in the valves to become plugged. Rusting may occur from presence of moisture. Premium grade hydraulic oils contain oxidation inhibitors and generally also rust inhibitors. Foaming can now be reduced to a negligible amount by certain types of additives.

INFLUENCE OF MOLECULAR STRUCTURE ON THE OXIDATION OF ALIPHATIC HYDROCARBONS, THE

by N. J. H. Small & A. R. Ubbelohde, Journal of Applied Chemistry, Vol. 3, Part 5, May 1953, pp. 193-198.

The object of this paper is to ascertain which of the various steps in hydrocarbon oxidation appear to be particularly sensitive to molecular structure. When the different steps in the reaction chain are considered separately, it appears most likely that the anomalous influence of structure on ease of oxidation is associated with the fate of the peroxide radical RO_2. One suggestion is that vibration coupling in an unbranched carbon-skeleton may favour the continuation of oxidation chains by increasing the activation energy available for a given bond-process.

INHIBITION OF CORROSION

by U. R. Evans, Chemistry & Industry, No. 22, May 30, 1953, pp. 530-533.

Inhibitors are mentioned (orthophosphates, carbonates, silicates, chromates, nitrates) which are much used for water employed in steel cooling systems, and are often grouped as "anodic" inhibitors; the alkaline compounds (phosphates, carbonates and silicates) provide ions (e.g. OH-) which can take part in the anodic reaction; the oxidizing agents (chromates and nitrates) probably act upon the metal exposed at sensitive spots and prevent or diminish anodic attach at such points. To some extent, intensification of attack can be avoided by using a cathodic inhibitor, which reduces the corrosion-rate by controlling the cathodic reaction. The best known is calcium bicarbonate. One of the most important of recent achievements is the use of benzoate inhibitors for treating the glycol-water mixture in engine-cooling systems where dissimilar metals are often in contact.

LITHIUM GREASES

by H. C. Meyer, Petroleum Engineer, Vol. 25, No. 5, May 1953, pp. C-15-16, C-18.

According to figures released by the Petroleum Administration for Defense, yearly production of lithium greases went from 54,123,732 in 1951 to 135,346,740 in 1953. The basic manufacturing procedures on lithium multi-purpose greases do not vary greatly from other types. The greases will contain from 5 to 10 per cent of lithium soaps. The unit operations involved are as follows: compounding, cooling, and milling. Although their manufacture is not completely free of difficulties, they do not present any problems that have not been previously encountered in the manufacture of other types of grease.

MECHANISM OF COMBUSTION CHAMBER DEPOSIT FORMATION WITH LEADED FUELS

by W. E. Newby & L. F. Dumont, Industrial & Engineering Chemistry, Vol. 45, No. 6, June 1953, pp. 1336-1342.

A study of the reaction mechanisms which result in the accumulation of deposits from fuels containing tetraethyl lead was carried out at the duPont laboratories. Gaseous lead oxide, produced from tetraethyl lead, does not undergo vapor state reactions except in the special case when the gas film near a clean combustion chamber surface is cool.

Under these special conditions it may be converted by hydrogen halides to gaseous lead halide. Solid lead oxide which condenses has a short life as a deposit constituent. It is attached by acid gases to form simple lead salts or reacts with the simple lead salts in solid state reactions to form complex lead oxy salts. The course of the reactions is dependent on the temperature of the combustion chamber walls. The reaction sequence is summarized for three different temperature conditions.

MINERAL OIL USED AS A HEAT TRANSFER MEDIUM

by B. M. Dunham & H. S. Ostrander, Canadian Chemical Processing, Vol. 37, No. 5, May 1953, pp. 66, 68.

The advantages of mineral oils as heat transfer media include low cost, ready availability, high stability and high thermal efficiency. When using mineral oils, every precaution must be taken to eliminate air from the system. It is also necessary to maintain the oil in the surge tank relatively cool in order to reduce oil oxidation to a minimum. Ready availability, low cost, low pressure operation, easy circulation, high stability and high thermal efficiency are advantages of their use.

NBS STUDY OF GAS VISCOSITIES

by Anon, A.S.T.M. Bulletin No. 190, May 1953, p. 35.

To provide data needed for the design of gas turbines and jet engines, the National Bureau of Standards is conducting a broad program of research on the viscosity of gases at elevated temperatures and pressures. Under the sponsorship of the nautics, hydrocarbon mixtures and combustion products composed of various proportions of nitrogen, carbon dioxide, argon, oxygen, carbon monoxide, and water vapor are being studied. In the course of this work, a unique type of flowmeter has been developed which accurately controls and measures the flow of very small volumes of gas. At present, available data on the viscosity of gases are very meager.

POLAROGRAPHIC STUDIES OF SULFUR COMPOUNDS IN PETROLEUM FRACTIONS

by M. E. Hall, Analytical Chemistry, Vol. 25, No. 4, April 1953, pp. 556-561.

Elemental sulfur, aliphatic and aromatic disulfides, tert-butyl trisulfide, and tert-butyl polysulfide give characteristic reduction waves with the dropping mercury electrode. Analytical methods, including electrolytes, solvents, and interferences, are discussed for each type when they exist in various mixtures. The most useful application of the polarograph has been the study of sweetening processes, where mercaptans are converted to disulfides and other types of sulfur compounds.

REGENERATION OF TRANSFORMER OIL

by Y. G. Emelyanov, Fuel Abstracts, Vol. 13, No. 1, Jan. 1953, p. 47.

Acidity curves show the satisfactory results obtained during operation over four years of thermo-siphon filters charged with silica gel attached to transformers to permit regeneration of oil in situ. Alternatively, a mobile filtration plant can be used without taking the transformer out of service, but the results are slightly less satisfactory. Before installing thermo-siphon filters, 18% of transformer oils in the Moscow area showed inorganic acidity.

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★ Filtering hydraulic oil eliminates sticking and clogging of valves and reduces down time and maintenance cost. A prominent authority on hydraulic machinery says that 70% of servicing and repairs is due to the improper condition of hydraulic oil.

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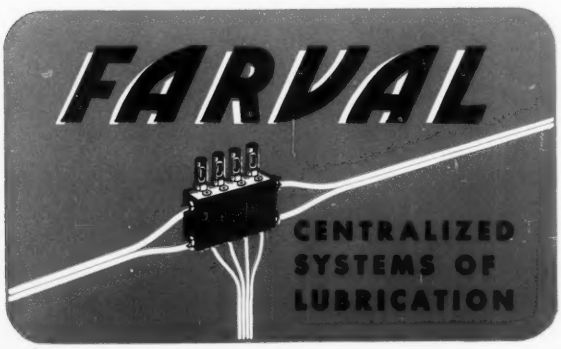

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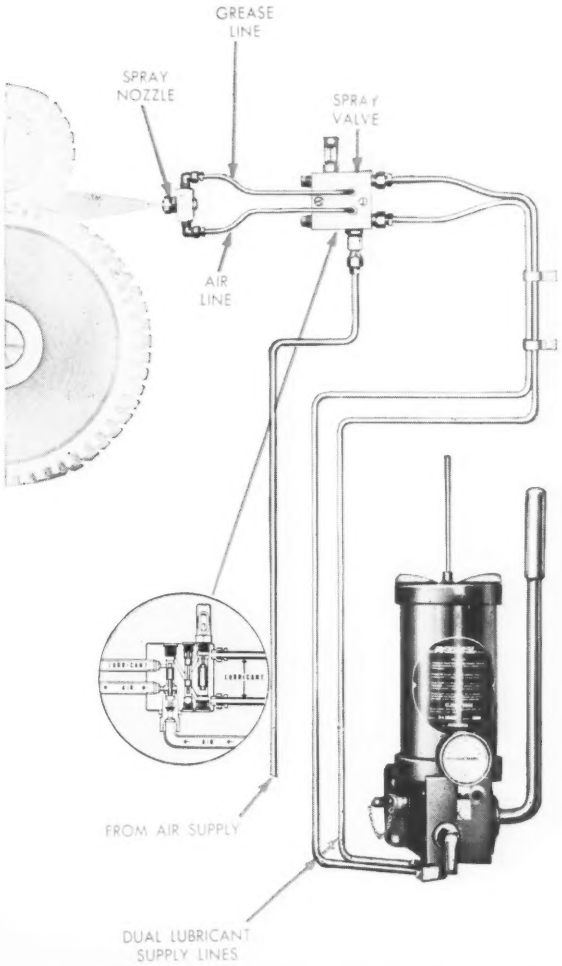

Drawing shows component parts of the Farval Spray Valve Centralized System of Lubrication—manual pumping station, dual lubricant lines, compressed air line, single Spray Valve and nozzle. Both automatic and manual systems installed three years ago are in operation today, prolonging gear life, saving labor and lubricant.

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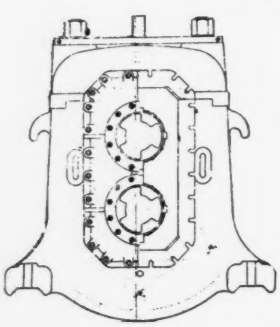

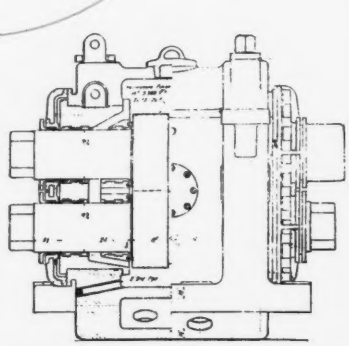

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